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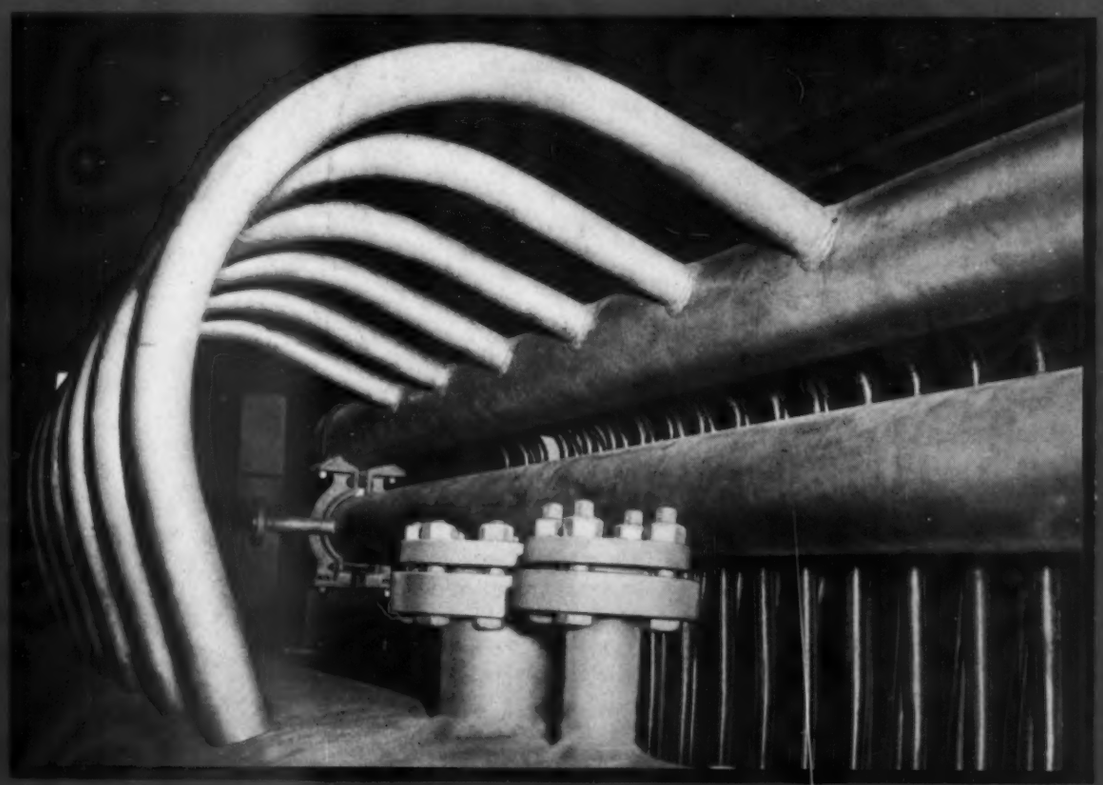


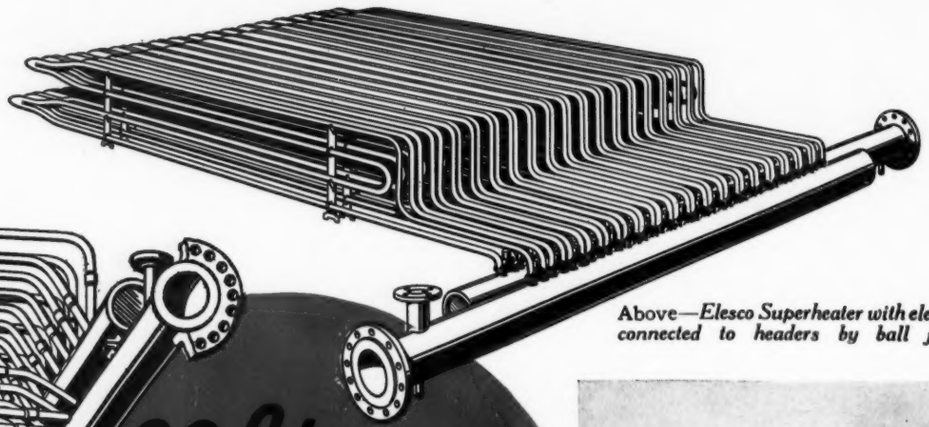
Tubes from boiler drum welded into superheater header

Superposed Turbine Blade Research

Superheater Arrangements

Combustion Calculations Using the Mol

Quality vs. Purity of Steam



Above—*Elesco Superheater with elements connected to headers by ball joints.*

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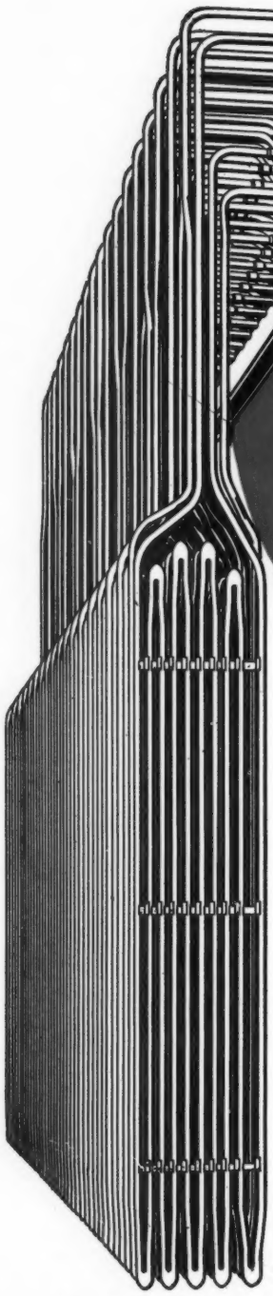
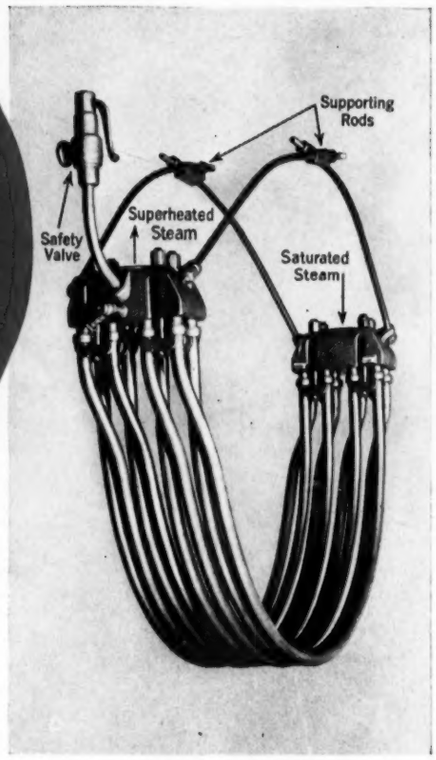


Elesco Superheater with elements connected to headers by welded joints.



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VOLUME ELEVEN NUMBER NINE

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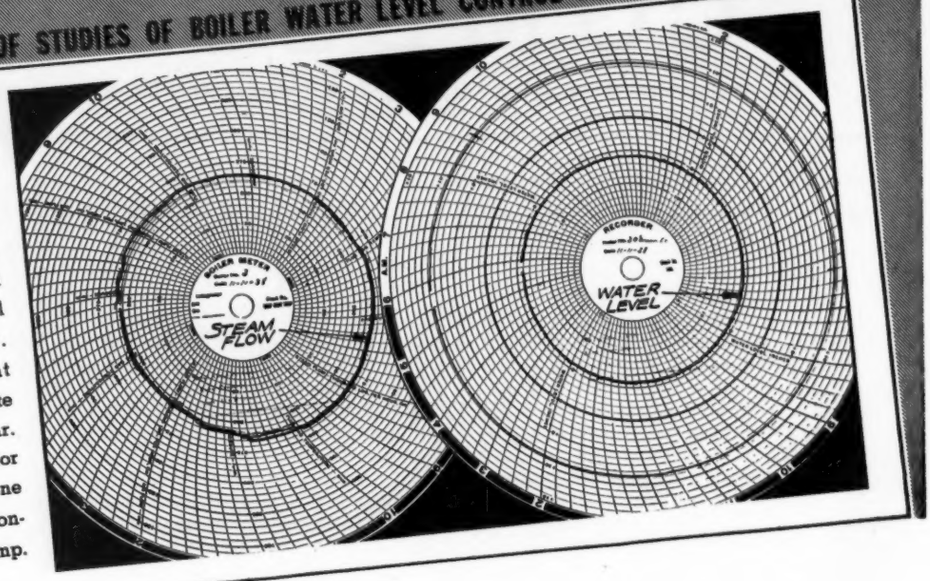

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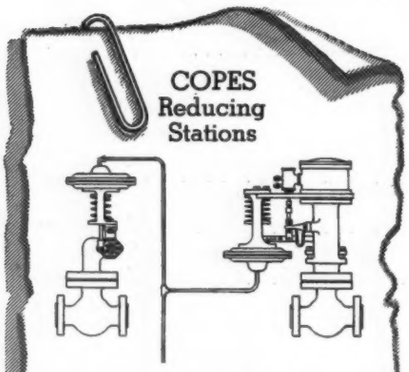

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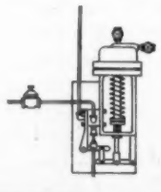

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EDITORIAL

Investigation Throws Light on Turbine-Blade Failures

The era of topping installations, employing high-pressure, high-temperature turbines operating at 3600 rpm, introduced a series of failures of first-stage impulse blades which interfered with operation of many of the machines under conditions for which they were intended. It was observed that these failures usually occurred during a very brief period of operation and were associated with partial loading of the machines, at which time the blades were subjected to shock imposed by intermittent admission of the high-density steam. Increasing the cross-section of the blades appeared to have little effect in alleviating the trouble and it became apparent that the failures were due to fatigue of the material.

To overcome this the admission valves of some machines were altered so as to change the distribution of steam entering the first-wheel buckets by admitting steam around the full periphery of the wheel—a procedure which slightly increased the steam rate of the turbine. In other cases efforts were made to operate the machines at near full load. But these measures were expedients and it was apparent that the solution rested upon further research. The program initiated by the Westinghouse Company, as described by Mr. Hague in this issue, was the result as far as one turbine manufacturer was concerned.

The investigations at the South Philadelphia Works under steam conditions of 400 pounds and 600 F total steam temperature served to prove that the blade fatigue was caused by shock and resonance which amplified the natural frequency of vibration of the blade material, and to indicate that the solution lay in the employment of material having a high damping effect or perhaps the use of separate damping devices. The subsequent investigations now under way on a full-size impulse stage, under actual operating conditions with steam at 1250 pounds and 900 F at Schuylkill Station, should be productive of further light on the subject.

Commemorating Science and Research

Protesting the defeatist attitude that invention and the machine are largely responsible for unemployment, 1500 persons on February 27 gathered at a testimonial dinner at the Waldorf-Astoria, New York, to honor over a hundred inventors, scientists and engineers as modern pioneers in raising the American standard of living. Those honored were all from the New York Metropolitan District, as the dinner climaxed fourteen similar regional meetings, under the auspices of the National Association of Manufacturers, in which more than five hundred were similarly honored. The occasion also marked the One Hundred and Fiftieth Anniversary of the American Patent System.

Throughout the speeches at the dinner ran the theme that modern frontiers of progress lay in the extension of industrial research and invention to the end that new products and new industries be evolved to absorb our labor surplus and further advance the standard of living.

It is significant that among those honored at New York there were fourteen individuals who had contributed to progress in the field of power. These contributions covered steam generation, combustion control, surface condensers, pumps, water treatment, bearings, transmission and power plant design such as to promote the production of power at reduced costs. This is a signal recognition of the role of power in modern progress.

It would seem that the country, as a whole, might be ready to turn once more to this basic concept of industrial progress, which until a few years ago was the dominant factor in the most amazing success story of modern times—the story of America's unparalleled advancement to the position of the world's leading industrial nation and the incidental attainment of the highest living standards ever known.

Coal Exports Stimulated

In normal times exports of coal from the United States are not large in relation to the total production; but conditions brought about by the war in Europe, and to a lesser extent by hostilities in the Orient, indicate that the present year will see a large increase in foreign shipments from this commodity. Figures for 1939 were about double those of 1938, the increase being largely in the last quarter of the year. Exports of coal to several South American countries showed a marked increase because of curtailment of their imports from Europe. Also China, which formerly utilized considerable tonnage from Indo-China, has had this source of supply interrupted by the Japanese and is now looking to this country to make up the deficiency.

With the recent attempted stoppage by the British of coal shipments from Germany to Italy, the latter, which must depend upon imported coal, will have to look elsewhere for its supply, unless the present controversy over such shipments yields to diplomatic negotiations. The British probably hope to secure this Italian market, but resentment over this interference may result in increased coal purchases from the United States.

In the aggregate these increased exports are unlikely to be of sufficient magnitude, in comparison with domestic consumption, to have any effect on coal prices in this country, but they should be of help to the coal industry, to employment among miners and to the railroads handling shipments to seaboard.

Superposed Turbine Blade Research*

By F. T. Hague, Engrg. Mgr., Westinghouse Elec. & Mfg. Co.

By means of mirrors placed in the impulse blades and in the hollow shaft, and a light beam, the vibration of the blade is recorded on a photographic film. Blade breakage was found to be due to fatigue and amplified stress resulting from resonance set up by the shock of dense steam impingement, superimposed on the natural period of vibration of the blade. This condition exists only at partial loads when intermittent steam admission is involved. The remedy appears to lie in proper damping. Following initial research at the Westinghouse Works, a full-size impulse wheel of 10,000 kw capacity has been set up to operate at 1250 lb, 900 F in the Schuylkill Station of the Philadelphia Electric Company, for further study under actual plant conditions.

SUPERPOSED turbines pass over a million pounds of steam per hour at a density five times as great as the low-pressure turbines with which they are in series and the steam temperature is around 900 F. The passage of a large volume of dense, high-temperature steam through the first wheel creates conditions unlike those with which there has been much previous experience. For each 20-deg increase in temperature in this region, the plastic flow or creep rate of steel is doubled and other physical characteristics are adversely affected.

To obtain good economy at partial loads the incoming steam is fed to this first wheel at only a portion of its periphery, and the blades passing into and out of this steam jet receive a most violent shock at each revolution. To form a mental picture of this shock imagine an impulse blade moving 350 miles an hour abruptly entering a steam jet density moving 1200 miles an hour. Shock and vibration of these first row blades is aggravated by this intermittent steam loading. This condition of intermittent steam loading has been conventional for many years on all sizes of condensing turbines. It has been only on those few machines of the large superposed class that its use, in an endeavor to develop the ultimate in partial load efficiency, has caused some blades to break at their root where they are fastened to the spindle. The prompt failure of three successive sets of physically larger blades on a large superposed turbine indicated that something more than brute strength is required of impulse blades in this application when intermittent steam loading is involved.

A research study was therefore initiated in order to develop a better understanding of the conditions affecting the operation of impulse blades in superposed turbines. Nothing short of actually photographing the blades' movements under actual operating conditions promised to give adequate and accurate records of the stresses imposed on these blades. To accomplish this an ingenious optical method, Fig. 1, was devised and the turbine blades are made to write their own story on a camera film adjacent to the turbine. Tiny mirrors are mounted inside eight different blades in the first and second blade rows. A light beam projected through a hole in the turbine shaft is reflected by an inclined mirror to the mirror in the whirling blade being photographed and then back out through the shaft to the stationary camera or viewing screen. By taking pictures every one-half second on a moving film, an exact record of blade performance is obtained for any desired set of operating conditions.

The first development turbine with this built-in optical system has been in operation at the South Philadelphia Works of the Westinghouse Company since last September. More than 20,000 photographic records have been obtained showing the performance of impulse blades in a wide range of operating conditions. Single blades and shrouded blade groups have been tested in the first and second rows of a Curtis stage under available plant steam conditions of 400 lb and 600 F. Though the pressure and temperature at our plant are not those met in commercial service, nevertheless the photographic method developed will apply with equal success to any commercial steam conditions.

The Evidence Obtained

Examination of a typical photographic record of a blade's movements during one complete revolution of the spindle shows that the blades vibrate like a tuning fork, as indicated in Fig. 2. The vibration is increased as a

* From a talk before the Philadelphia Section, A. S. M. E., on February 27, following an inspection of the test turbine at Schuylkill Station.

blade enters the steam jet, it vibrates while in the jet, and then continues to vibrate with slowly decreasing amplitude after it leaves the jet. This slowly decreasing amplitude of vibration is significant because it is caused by the internal molecular friction of the blade material which will hereafter be referred to as the damping characteristic of the blade steel. The blades vibrate at their natural frequency. This is true whether the steam shock is light or strong and is independent of turbine speed. The

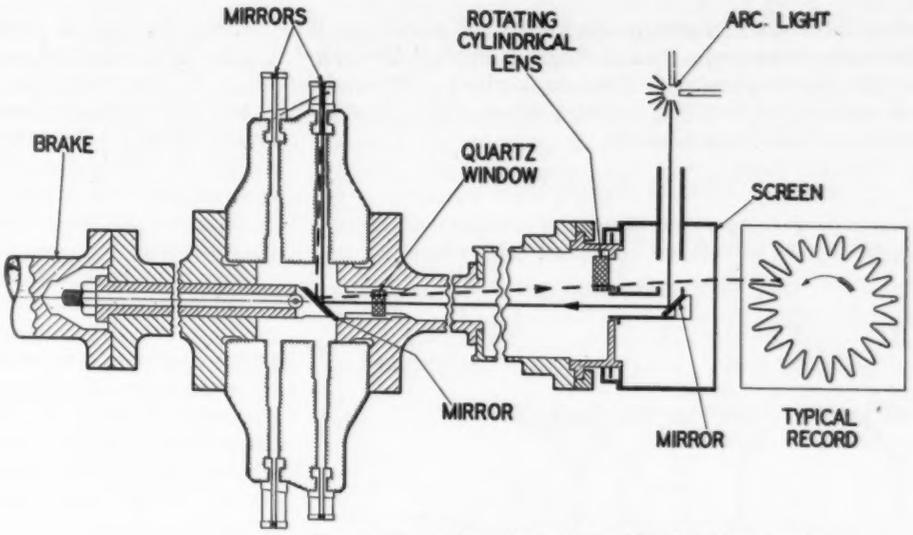


Fig. 1—Optical system of impulse-blade research

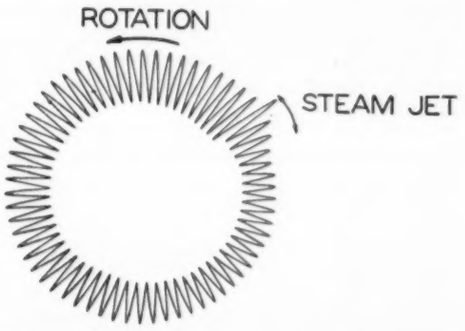


Fig. 2 —Typical blade vibration in resonance

natural frequency of vibration is fixed by the blade design and by the material.

In superposed turbines the frequency of vibration of first-row blades is about 2500 cycles per second, corresponding to one of the highest notes on a piano, or 200,000,000 cycles during a 24-hr day. Blades stressed at their fatigue limit usually break within 10,000,000 cycles. This explains why failures of blades in the first row of superposed turbines usually develop quite promptly, if they fail at all, particularly if the turbine is loaded at half load, or below, where intermittent steam loading is involved. As the turbine load is increased the steam jets cover more and more of the periphery of the first

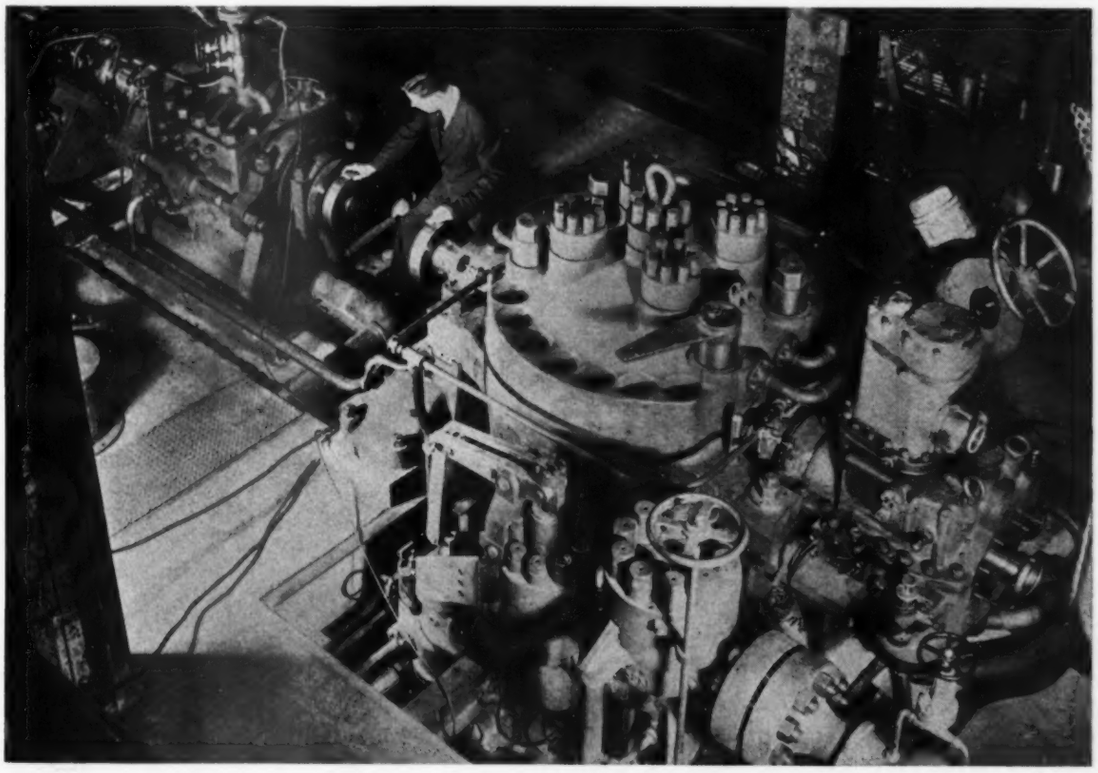


Fig. 3—High-pressure experimental unit of 10,000 kw capacity at Schuylkill Station of Philadelphia Electric Company

wheel until above three-quarters load full admission is ordinarily employed; that is, steam is applied throughout the entire periphery of the first wheel. When the full admission condition is established, this vibratory stress condition is eliminated.

What the Studies Show

As in diagnosing a new ailment, it is important to know which factors affect the condition and which do not.

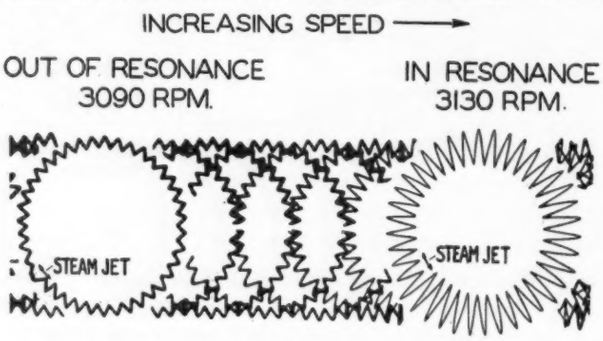


Fig. 4—Record of blade motion at constant load

The optical test studies and practical experience have shown both.

Strange though it might seem, the amount of load on the turbine has relatively little to do with the maximum blade stress. The factors which influence this condition are mechanical resonance and the damping characteristic of the blade steel.

Mechanical resonance involves a relationship of two frequencies. In turbines one of these is the frequency of the shock from the steam jet; the other is the natural vibrating period of the blade. When these two frequencies are in such a relation that one is an exact integral multiple of the other, resonance results. To illustrate this, imagine a slender steel rod with a ball at its top as representing a turbine blade. By tapping the ball with a wooden hammer we can represent the shock from the steam jet. If the repeated tapping of the ball

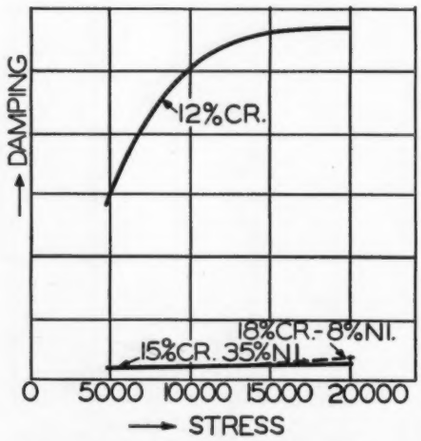


Fig. 5—Damping of blade steels at 700 F, 750 cycles per sec

be in phase with the existing vibration an amplitude of large value can be built up. This condition is called *resonance.* If, however, the frequency of tapping be changed and the ball be hit out of phase with the vibration, a new order of motion of much smaller magnitude is set up. This is *out of resonance.*

Every mechanical system has a natural period of vibration and every one can be set into resonance by being hit on the right frequency. When resonance is produced in any mechanical system, the magnification of stress is limited solely by the damping of the material used. The ability of a turbine blade to operate without breakage depends largely on whether the damping within the blade is sufficient to choke the vibration established during resonant conditions.

With any constant turbine load, the magnitude of blade stress has been found to vary over a four-to-one range as the turbine speed is changed as little as 2 per cent. It is this four-to-one range or build-up of blade stress at constant turbine load which is the cause of the breakage of blades in superposed turbines. Fig. 4 is taken from an actual turbine in constant load operation wherein the only variable is a slight change in turbine speed. In this particular test turbine it took a change in speed of $1^1/_3$ per cent to go from the resonant condition to the non-resonant condition. "Out of resonance" is illustrated at the left of the figure and "in resonance" at the right. This represents seven or eight photographic records of blade performance taken one-half second apart as the turbine speed is slowly changed from the 3090

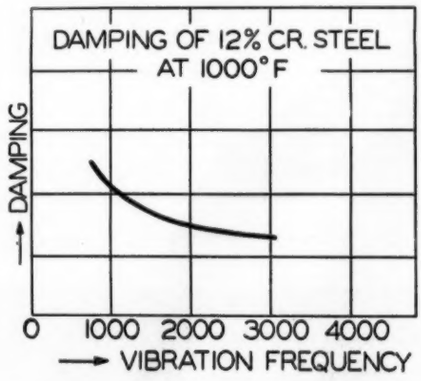


Fig. 6—Damping of 12-per cent chrome steel at 1000 F

rpm to 3130 rpm. This variation of speed is the only difference between these successive records of blade performance. In order to get all of these records on one chart to show them in sufficient scale these records were taken overlapping and for the sake of clarity all of the overlapping records have been rubbed out on the first and last ones. They represent the "out of resonance" and the "in resonance" blade performance records.

Refer to the "out of resonance" record when the blade passes the steam jet. The steam jet hits that blade out of phase with its existing motion and one can see the new order of frequency that is established. It is that continual hitting of the blade out of phase with its existing motion as it passes through the steam jet that prevents the blade from building up its vibration to a large amplitude and building up its stress to a large value.

In contrast to this examine the record on the extreme right and notice that as the blade passes through the steam jet, there is no variation whatsoever in the frequency. In that case there are established the conditions required for resonance. The steam jet is hitting the blade in phase with its existing vibration and, as a result, it builds this vibration amplitude up to a large value, controlled and limited only by the damping ca-

pacity of the blade steel. It is that condition which increases the blade stress by a factor of four to one, even though the turbine load is constant, and it is that amplification of blade stress which breaks the blades in superposed turbines.

It is obvious, therefore, that the impulse blading of large superposed turbines must be designed to operate at safe stresses under resonant conditions. While actual turbines run at constant speed the natural period of blades will vary more than 10 per cent by virtue of manufacturing variations, changes in operating temperatures, etc., so that some of the blades will be in resonance under all operating conditions.

This four-to-one range of amplification of blade stress at constant load would be increased to between 10 and 15 to 1 if a chrome-nickel or nickel-chrome steel were used instead of 12-per cent chrome steel, which has greater damping effect than any other steel we have tested. Conversely, if the damping capacity of this blade were to be increased ten times by the use of a separate damper, that amplification of blade stress could be reduced to two to one, or less, all of which simply emphasizes that damping is the factor of safety of impulse blades.

Damping

One phase of our investigation has been to explore the factors affecting damping, and if, under proper conditions, the inherent ability of a blade to protect itself is enough, or if some additional help must be administered. Several important damping relationships are shown.

1. Damping is not constant but increases with blade stress and is extremely low at low stress (see Fig. 5). It is as important to know this relationship as to know the strength of steel.
2. The damping characteristics of different steels vary greatly. The widely used 12 per cent chrome steel has more damping in its usable application range than any other material yet investigated. The materials of the chrome-nickel class that excel in high strength at high temperatures have about one-tenth as much damping as 12 per cent chrome steel.
3. Damping becomes less as the frequency of vibration increases as is indicated in Fig. 6. This means that the increased stiffness of heavier blades does not result in much lower blade stresses.
4. Damping decreases rapidly at higher operating temperatures, particularly beyond 750 F. At 1000 F the damping is only one-third as much as at room temperature, and is decreasing precipitously (see Fig. 7).

Separate Dampers

These findings suggest the possibilities of applying some sort of separate dampers to the blades. We are investigating separate damping means that increase the damping of blades by five to ten times. A steel capsule, partly filled with material that is fluid at operating temperatures can be inserted inside a blade and maintain its full damping capacity up to 1500 F.

It is not meant to infer that superposed turbines for present-day standards of temperature and pressure must have separate blade dampers in order to operate safely. This research merely points out that such means would greatly increase the factor of safety of impulse blades and such means may be required to meet future conditions.

Dampers are not unusual devices. A great many rotating mechanical systems require them. Automobiles have them. Diesel engines of any considerable size wouldn't run without them. The modern airplane engine, particularly in the high-power fighting ships and bombers, has as many as three separate dampers and if it weren't for these dampers, the engines couldn't develop over half their power.

Calculation of Blade Stresses

Our research shows that impulse-blade stresses can be calculated. The calculation system that has been developed, based on this resonant method of analysis, permits a rational evaluation of controllable design features and makes it possible to predict whether a certain design is safe.

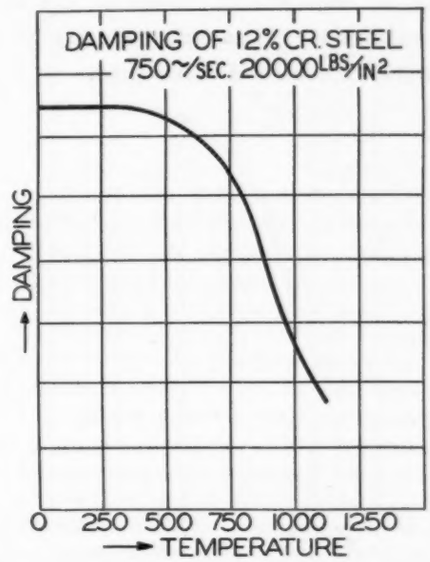


Fig. 7—Damping of 12-per cent chrome steel at 20,000 lb per sq in. and 750 cycles per sec

Building on the experience with the experimental turbine at the South Philadelphia Works, a second unit with full-size impulse wheel, a duplicate of the impulse stage of the 50,000-kw West Penn unit, was built for research study of blade performance under 1250 lb 900 F steam operating conditions. This turbine with its 10,000-kw generator and 15,000-hp water brake has been set up for testing at the Schuylkill Station of the Philadelphia Electric Company. It takes steam from the station 1250-lb steam lines and exhausts into the low-pressure 240-lb lines permitting its operation as a superposed unit without interference with the station's regular operation. It is built solely for research purposes and has many special constructional features, including single and double blades, separate blade dampers, different shroud attachments and various blade groupings. The optical system will record the performance of sixteen different blade combinations. Two rotors are being built with different blade designs and the presence of the 10,000-kw generator will permit life tests by loading into the company's system.

Reference to this unit must acknowledge the joint research relationship with the Philadelphia Electric Company which is making available the use of its station, operators and steam supply.

SUPERHEATER ARRANGEMENTS

By FREDERIC I. EPLEY
Combustion Engineering Company, Inc.

Typical arrangements for convection type superheaters are shown to suit different designs of boilers of small, medium and large capacities and for different ranges of superheat. Each is discussed with reference to its adaptability to the operating conditions to be met.

ONE encounters a wide variety of superheater arrangements, the selection depending upon the conditions surrounding the individual boiler installation. Among the factors that influence superheater location are the type of boiler, the permissible draft loss, the character of the fuel, desired accessibility for cleaning, the total steam temperature and its control over a specified range in load; the higher the steam temperature, the greater the need for close control.

In this connection it is well to bear in mind that with a convection-type superheater the temperature tends to increase with the load, due to the greater mass flow of the products of combustion. That is, if the superheater be designed to produce a given steam temperature at a specified load, the temperature may be too high at greater steam outputs or too low at lesser outputs. This can be offset to some extent by location of the superheater and to a very large degree by providing bypass dampers for controlling the path of the gases. Such damper control is now quite generally employed for high-pressure, high-temperature steam generating units, but to a lesser extent on those for lower steam temperatures under around 800 or 825 F. Furthermore, the slagging conditions of the furnace walls and boiler convection surface ahead of the superheater, during operation, will affect the heat absorption by those surfaces and consequently the temperature of the gases entering the superheater, which, in turn, affects the total steam temperature.

Generally speaking, convection superheaters are classed, by location, as "interdeck," "overdeck," "interbank" or "intertube." In box-header or straight-tube sectional-header boilers the superheater may be placed either above the main tube bank, between the first and second passes, in which case it is designated as "overdeck"; or in the first pass between the upper and lower banks of tubes, which position is known as "interdeck." In such boilers the superheater elements are usually parallel to the boiler tubes, except in the case of the interdeck arrangement in box-header boilers where they are placed at right angles to the tubes to facilitate removal. With bent-tube boilers the superheater may be located between the first and second banks of tubes (interbank) or between the tubes of the first pass (intertube). The former arrangement is better adapted to superheat control and is preferable with slagging coals, although the latter arrangement makes possible somewhat less superheating surface, as the elements are exposed to hotter gases and also receive some radiant heat from the furnace. There are other arrangements, dictated by the design of boiler, in which the superheater elements are placed crosswise of the tubes of a bent-tube boiler.

The accompanying sketches indicate a variety of superheater arrangements as applied to different types of boilers and for different conditions to be met. These will be discussed with reference to the adaptability in each case.

Application to Return-Tubular Boilers

The simplest case is perhaps the return-tubular boiler in which a superheater of the girth type is applied around the lower half of the shell, as shown in Fig. 1. Such superheaters are designed to fit the standard shell diameters of h r t boilers. The degree of superheat attainable in such cases depends upon a number of operating factors. Usually about 80 deg of superheat is attainable, although as high as 125 deg has been obtained under favorable conditions. Where a higher degree of superheat is desired, or when a minimum variation in total steam temperature is required with this type of boiler, the superheater elements are suspended in the rear combustion chamber.

Arrangements for Horizontal Straight-Tube Boilers

Fig. 2 represents a longitudinal-drum, box-header boiler with cross-baffling in which the superheater is located above the boiler tubes; that is, overdeck. The same location would apply to a cross-drum boiler of this type, or to a sectional-header boiler, regardless of the make. The degree of superheat that can be obtained depends not only upon the amount of superheater surface that can be installed in the space available, but also upon operating conditions, such as type of firing, fuel, excess air, saturated steam quality and water cooling in the furnace. A rather large superheater is required to obtain as much as 250 deg of superheat, even under favorable operating conditions.

When higher steam temperatures are required with a straight-tube boiler, it becomes necessary to locate the superheater in an interdeck position as indicated in Fig. 3, in order to expose the elements to hotter gases; the higher the temperature required, the fewer the boiler tubes between the furnace and the superheater. In such a location it is possible to provide damper control, as indicated, in order that part of the gases may bypass the

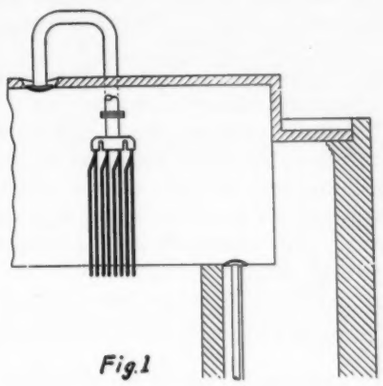


Fig. 1

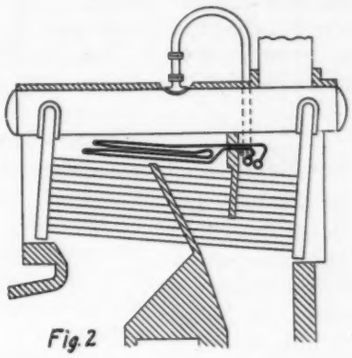


Fig. 2

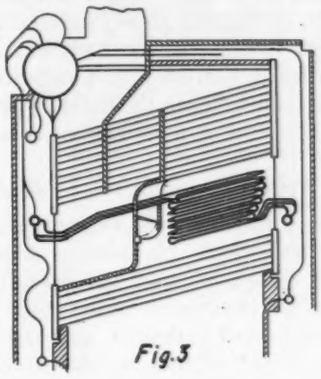


Fig. 3

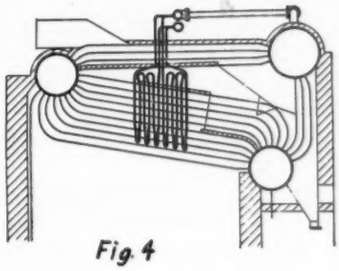


Fig. 4

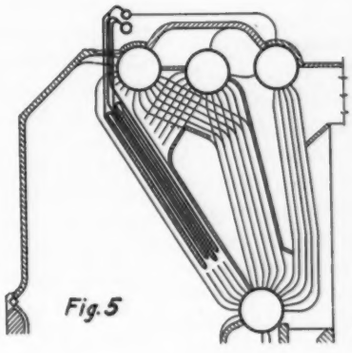


Fig. 5

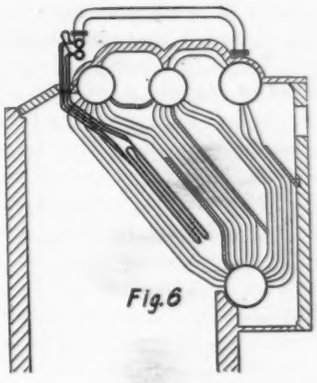


Fig. 6

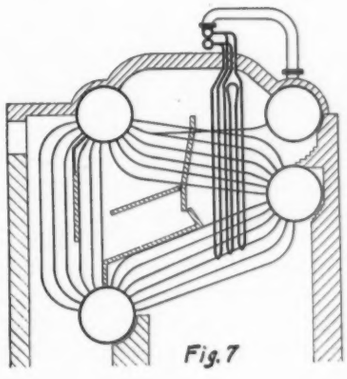


Fig. 7

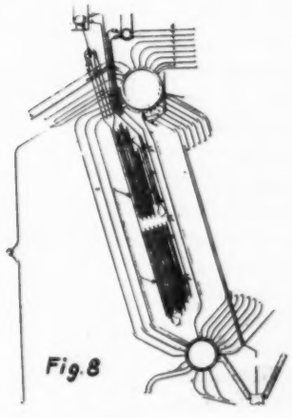


Fig. 8

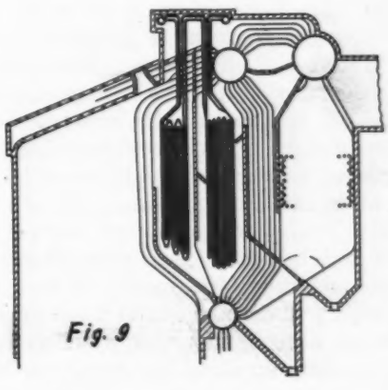


Fig. 9

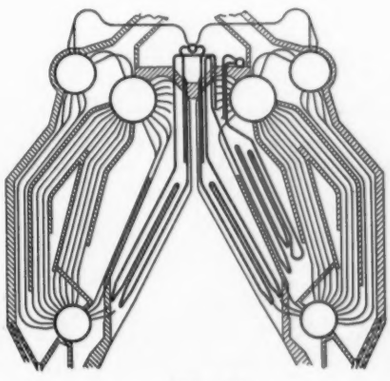


Fig. 10

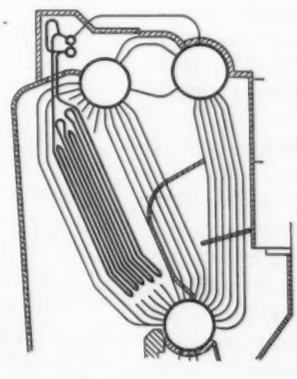


Fig. 11

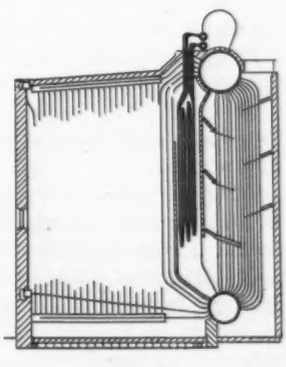


Fig. 12

Figs. 1 to 12—Typical superheater applications

superheater at high loads or when the furnace walls are covered with slag. Steam temperatures of the order of 900 F, or slightly more, may be obtained with this arrangement. In the illustration, it will be noted that the superheater headers are outside the boiler setting, but it is possible to locate the headers inside the setting if they are protected by a water-cooled baffle.

Applications to Multi-Drum Bent-Tube Boilers

Fig. 4 shows a type of superheater that has been applied successfully to a three-drum low-head design of boiler. Lack of space makes it impossible to employ a horizontal interbank superheater, hence the elements are suspended vertically and extend between the boiler tubes, which cannot be staggered. With this arrangement total steam temperatures up to 700 F are attainable under favorable conditions.

Figs. 5 and 6 represent typical superheater applications to four-drum boilers of the bent-tube type. In the former the superheater elements are placed intertube across the first bank of boiler tubes, and in the latter the superheater is back of the first bank (interbank). In both cases, however, they are in the first pass. In the intertube arrangement, Fig. 5, the superheater elements are supported directly from the boiler tubes by means of high-heat-resisting alloy castings. A considerable amount of heat is absorbed by radiation directly from the furnace which tends to reduce the variation of steam temperature over a range of boiler ratings by combining the effects of radiant and convection heat transfer. Furthermore, by thus placing the superheater elements in the space between the boiler tubes, the front bank of the boiler is made more effective by breaking up the straight flow of the gases through the bank. On the other hand, this intertube arrangement is not as desirable where the fuel has slagging characteristics, as the front boiler bank becomes more susceptible to slagging and the slag is more difficult to remove. Under such conditions the interbank arrangement, as shown in Fig. 6 is preferable. Furthermore, bypass damper control is applicable to interbank superheaters whereas it is not to those of the intertube type, except in the case of the double-set boiler, as discussed later in connection with Fig. 10.

In the design of four-drum boiler shown in Fig. 7 the superheater elements are hung vertically in the first pass between and almost at right angles to the tubes of the first and second boiler bank. This rather unusual placing of the superheater is dictated by the boiler design. It will be noted that the gas flow is parallel to the elements as compared with the combination of parallel and transverse flow, indicated in Figs. 5 and 6. A slightly greater heat transfer rate is obtained with cross-flow of the gases.

By placing a baffle in front of the superheater, as in Fig. 8, it is possible to control the gas flow over the superheater elements more closely than in the arrangements shown in Figs. 5 and 6. Such a design is employed primarily in large high-pressure, high-temperature installations where the control of steam temperature is desirable over a considerable range of evaporation, or where large amounts of furnace water-cooled surface are employed. It will be noted that there is a bypass damper located at the top of the second baffle by means of which the amount of gas passing over the superheater elements can be controlled. The relation between gas temperature drop and steam temperature rise is approximately the same as in a counterflow heat exchanger. In some installations the damper is located in a lower gas temperature zone.

The design shown in Fig. 9 represents a three-drum boiler with a two-pass superheater, arranged for high superheat, in which the front or high-temperature section has the elements on wide centers to minimize slagging and permit easy cleaning, and the rear section which is in a lower temperature zone has about half the spacing of the front elements. The materials of the elements are made suitable to withstand the temperatures in the respective gas zones. With large superheaters such as this it is much easier to support them in a vertical position.

Fig. 10 represents a design in which the principle of the compensating superheater is employed. This has been used successfully on a number of units. The gas flow at maximum evaporation is divided equally between the two sides of the boiler. As the evaporation decreases, a larger portion of the gases is made to pass, by means of a bypass damper, through the right-hand side which contains the larger superheater and has both intertube and interbank sections. By this means a very flat superheat curve is obtained over a wide range in load.

Many times it is required to operate a superheater initially at relatively low steam temperature with the intention of employing higher superheat at some future time. This may be accomplished by the arrangement shown in Fig. 11 with only a portion of the elements installed across the width of the boiler. However, in order to maintain a reasonable pressure drop through the superheater, which has only part of its elements installed, it is necessary to place jumpers, or bypass loops, between the saturated and the superheated steam headers, as shown. It is desirable, however, to keep the number of these bypass loops at a minimum in order to maintain adequate steam velocity through the superheater and thus protect the elements from burning.

The two-drum bent-tube boiler is now being employed extensively in industrial plants and to some extent for central stations, in capacities ranging from about 20,000 to 200,000 lb per hr or slightly over, pressures up to 900 lb per sq in. and total steam temperatures up to 850 F. There are several makes of boilers embodying this general design of which Fig. 12 may be taken as typical. Here the superheater is of the interbank type, and suspended vertically between baffles. Distribution of gas flow over the superheater elements is good and a bypass damper may be placed at the top of the second baffle where conditions make such control desirable.

While the foregoing represent typical arrangements of convection superheaters for stationary units of small, medium and large capacities and for steam temperatures from 450 F to over 900 F, there are also many modifications of these, too numerous to include here, to suit special boiler designs and unusual operating conditions. In each, however, the superheater is an integral part of the steam generating unit—a practice that is general where the steam is employed for power production.

Separately-Fired Superheaters

In some cases where the steam is used for process, very close control of its temperature is imperative. Under such conditions a separately-fired superheater may be

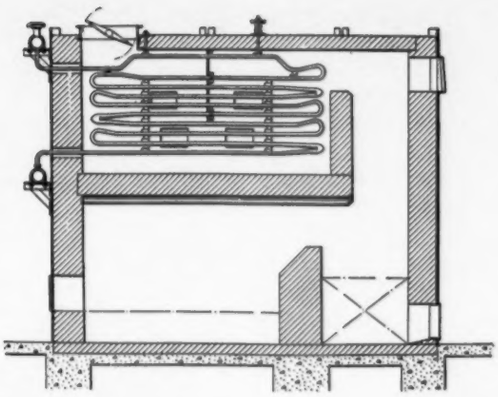


Fig. 13—Combination waste-heat and direct-fired superheater

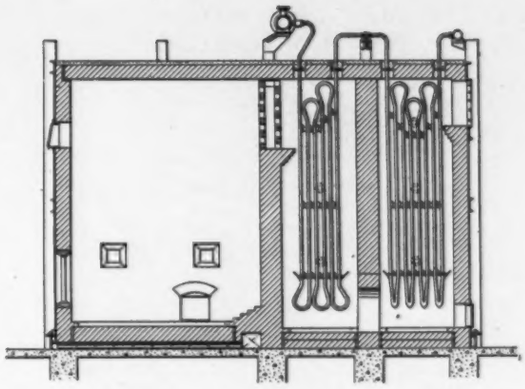


Fig. 14—High-temperature oil- or gas-fired superheater

employed. These may be directly fired with coal, oil, gas or waste heat and designed for capacities varying from less than 500 lb of steam per hour to as high as 300,000 lb per hr. Temperatures range from 450 to 1200 F. Automatic control is frequently applied. The arrangements employed with such units are quite numerous, but Figs. 13 and 14 may be considered representative. The former is a combination waste-heat and direct-fired layout and the latter a high-temperature design for oil or gas firing.

Separately-fired superheaters have also been used in connection with existing power boilers of such designs as to make the installation of an integral superheater uneconomical. Such installations are generally in connection with old discontinued designs of boilers which are kept in good operating condition.

Small separately-fired superheaters are also designed for use in test work and frequently cover a wide range of operating conditions.

Fly-Ash Collectors

The recent report of the Prime Movers Committee, E.E.I., on "Combustion" comments on the increasing use of fly-ash collectors of both the mechanical and the electrostatic types and their continuous improvement.

Ten years ago a dust-removal efficiency of 60 per cent would have been about all that could be expected, whereas in test data received for this reports efficiencies, of over 90 per cent have been recorded.

While the primary purpose of the fly-ash collector is the prevention of atmospheric nuisances in industrial and residential districts adjacent to power plants, other factors which may be listed under power plant economies include (1) a reduction of fly-ash erosion of the induced-draft fan blading, (2) a reduction in power plant cleaning costs and (3) lessening of short-circuit hazard in sub-stations.

In densely populated regions coal must be shipped by rail or water; consequently, as freight charges go up, ash content of the coal comes down, for it is poor economics to pay freight on ash or moisture. In other words, in places remote from coal mines the fuel purchased is generally high in heat value and low in moisture and ash. In such cases the reduced fan maintenance, increased cleanliness and lesser short-circuit hazard would probably fall short of establishing economic justification for the installation of fly-ash recovery equipment and its cost must be charged against betterment of public relations.

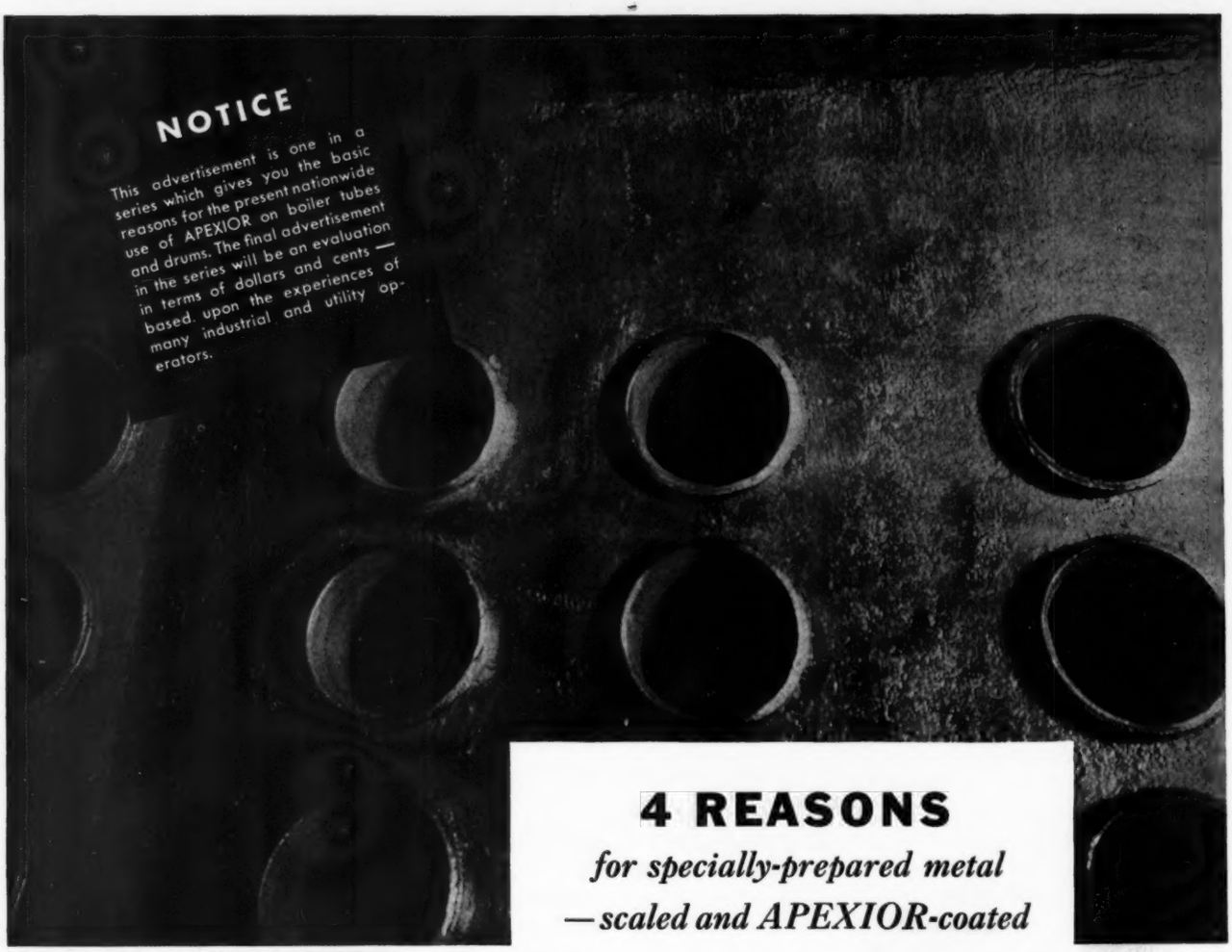
On the other hand, where a power plant is located near or in a coal-mining district the use of low-grade coal is usually justified. With an ash content above 10 per cent, and especially if the ash has high abrasive properties, replacement cost of fan blading in pulverized-coal-fired installations may be sufficiently high to compensate for the cost of a fly-ash collector.

With pulverized-coal-fired boilers of large capacity and operated at high ratings about half the ash is drawn out of the boiler in the form of fly ash or dust. These conditions have encouraged intensive study of various types of collectors now on the market. Each type has somewhat different operating characteristics. For example, the most efficient operating range of the electric precipitator occurs when handling the fine grades of fly ash or the smaller particles of dust, whereas the cyclonic type operates most efficiently on the relatively coarser ash particles. Unburned carbon in the flue gas adversely affects the performance of both types, although to a somewhat greater extent in the case of the electric precipitator.

The successful operation of the electric precipitator depends to a very important extent on a uniform distribution of the flue gas to the inlet of the unit. To obtain this result, various devices have been developed, including guide vanes in the ducts and perforated plates at the precipitator inlet or, in some cases, a combination of both.

Where removal of sulphur from the flue gases is considered desirable, the flue-gas washer can be used for this purpose and at the same time for fly-ash removal. Several types of such scrubbers are in successful operation with reported satisfactory collection efficiencies, but maintenance of the scrubber is somewhat higher than for other collectors, because of the corrosive nature of the material handled.

Company statements to the committee indicate that maintenance of all types of collectors, in general, is not excessive and no unusual boiler outage can be charged to this equipment.


NOTICE
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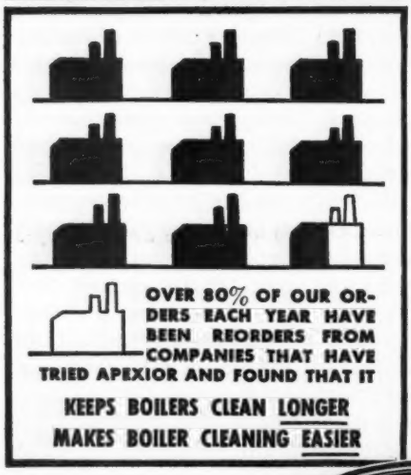
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Free Fusible Material in Coal Ash

This investigation was in two steps, the first dealing with coal crushed to 4-mesh size to simulate stoker firing, and the second with samples ground to pass through a 200 mesh and then separated into gravity fractions to cover firing in pulverized form. Data showing the proportions of the total ash in different ranges of initial deformation temperature explain the occurrence of slagging at temperatures well below those indicated for the coal ash, as a whole, as determined by the conventional method.

AN INDEX OF CLINKER AND SLAG FORMATION*

By G. B. GOULD†
and H. L. BRUNJES†

IT has long been recognized that the ash-softening temperature, as determined in the laboratory, had one weakness in that it represented the temperature at which a thorough mixture of the whole ash of the coal would melt, the mixing having taken place before the heat was applied. This does not, of course, simulate what takes place in practice.

The present investigation started from the observation that some clinkering and slagging obviously take place in a boiler furnace at temperatures below the laboratory-determined ash-softening temperature of the coal. Knowing that coal as shipped is a heterogeneous mixture of diverse mineral substances, each of which may act independently in the initial stages of clinker and slag formation, it seemed logical to investigate separately the softening temperatures of the ash in physically separable portions of coal.

Two methods of investigation were used; one aimed to throw light on the formation of clinker in a solid fuel bed; the other relating to slag formation in a pulverized-coal furnace. Identification of the coals studied and analysis of the head samples are given in Table 1.

TABLE 1—IDENTIFICATION OF COALS AND ANALYSES OF HEAD SAMPLES

Coal	Seam	County	State	Analysis (Dry Basis), Per Cent Volatile	Ash	Sulphur	British Thermal Units	Initial Deformation Temperature, Deg. F	Ash-softening Temperature, Deg. F
A	Lower Freeport	Jefferson	Pa.	28.0	9.9	0.8	13,760	2800+	2800+
B	Pittsburgh	Westmoreland	Pa.	29.7	8.4	0.9	14,295	2780	2800+
C	Upper and Lower Freeport	Indiana	Pa.	31.2	6.3	0.7	14,535	2600	2700
D	Upper Freeport	Armstrong	Pa.	31.8	9.3	1.2	13,635	2480	2580
E	Lower Freeport	Clearfield	Pa.	29.9	8.5	1.2	14,140	2400	2480
F	Thick Freeport	Allegheny	Pa.	35.9	6.5	1.1	14,370	2360	2460
G	Pittsburgh	Monongalia	W. Va.	35.8	7.3	2.1	13,965	2080	2140

Relation to Clinker Formation

In the study related to clinker formation, each coal was reduced to approximately the top size, from 95 to 100 per cent through a 4-mesh screen. Either $1^1/_4$ or 2-in. nut slack was used. The extent to which coal is crushed in a stoker varies with the friability of the coal and the type of stoker, but there is no question that considerable crushing action takes place.

In order to secure a maximum of differentiation of materials, each sample, after crushing, was separated into four gravity fractions, and each of these into six or seven size groups. This resulted in 24 or 28 portions of each sample separated according to the size and the density of the particles. Each portion was then ground to 60 mesh and tested for the percentage of ash and the ash-softening temperature. By this procedure it became possible to determine the fusibility of the ash by classes of material, which are capable of acting independently in the fuel bed, and also the proportion of the whole ash in the coal which is contributed by each portion. It also throws light on the distribution of the fusible material.

Fusibility of Ash in Separable Portions of Crushed Nut Slack

The data for coals *A*, *B*, *C*, *D*, *E* and *F*, as given in Table 2, show that coals having substantially the same ash-softening temperature, as determined in the conventional way, have strikingly different proportions of fusible material and significant differences in its distribution. Initial deformation temperature is given instead of ash-softening temperature for the individual fractions, because the study seemed to emphasize the importance of the independent action of the ash in separable portions of the coal in starting clinker formation.

While both coals *A* and *B* would be rated at 2800 F by the standard laboratory method, it will be noted that coal *B* contains more than twice as much fusible ash so distributed as to be capable of independent action in starting a clinker. Coal *C*, while rated at 2700 F, has a third more fusible material in a free state than coal *B*.

* From a paper before the American Institute of Mining & Metallurgical Engineers, New York, N. Y., February 14, 1940.

† President and Fuel Technologist, respectively, of Fuel Engineering Company of New York.

TABLE 2—PROPORTIONS OF TOTAL ASH IN EACH RANGE OF 100 DEG INITIAL DEFORMATION TEMPERATURE FOR COAL CRUSHED TO 4 MESH

Coal	Head Sample A.S.T. Deg F	Under 2100°	2100°–2199°	2200°–2299°	2300°–2399°	2400°–2499°	2500°–2599°	2600°–2699°	2700°–2799°	2800° and Over
A	2800+	..	..	8.2	...	5.0	2.6	16.3	26.7	41.2
B	2800+	2.3	11.1	4.9	...	1.2	...	..	..	80.5
C	2700	..	24.3	...	2.5	...	...	8.2	..	65.0
D	2580	26.3	11.9	...	...	2.5	1.9	5.0	5.5	46.9
E	2480	9.1	22.5	9.5	4.7	..	2.3	1.5	21.3	29.1
F	2460	27.8	4.1	...	...	2.2	3.7	6.8	28.2	27.2

Table 2 shows that coal *D* has a much larger proportion of the ash having an initial deformation temperature above 2800 F, which probably accounts for the higher average ash-softening temperature, but this may have no practical effect on clinker formation, when the proportion of fusible material, which would start clinker formation, is very nearly the same in both coals. The difference between coals *D* and *F* is even more striking. Coal *D*, with an ash-softening temperature 120 deg higher, contains a substantially greater proportion of free fusible material.

tions, according to the distance from the burner, and thus produces a variety of slags having different physical properties. Slag removed from superheater tubes is often found to be composed almost entirely of refractory material bound by a very small proportion of fusible ash.

The conclusion, therefore, would seem to be that the most important factors in predetermining the slagging characteristics of a coal of this general type in a pulverized-coal furnace is the *amount of free fusible material*, and the relative amounts of fusible and refractory materials.

TABLE 3—PROPORTIONS OF COAL AND ASH, AND INITIAL DEFORMATION TEMPERATURE FOR EACH GRAVITY FRACTION, 200-MESH COAL

	A			*B*			*C*			*E*			*F*		
	Coal, Per cent	Whole Ash, Per cent	I.D.T., Deg F	Coal, Per cent	Whole Ash, Per cent	I.D.T., Deg F	Coal, Per cent	Whole Ash, Per cent	I.D.T., Deg F	Coal, Per cent	Whole Ash, Per cent	I.D.T., Deg F	Coal, Per cent	Whole Ash, Per cent	I.D.T., Deg F
Float 1.3	4.9	1.0	2560	24.7	8.3	2800+	16.3	5.4	2440	35.0	9.9	2640	34.3	12.6	2600
1.4	73.7	39.8	2800+	62.0	46.8	2800+	72.3	53.2	2800+	49.8	37.6	2800+	56.1	37.3	2780
1.5	12.1	19.2	2800+	7.6	16.7	2800+	6.3	11.9	2800+	5.5	7.0	2800+	3.7	9.5	2800+
1.6	3.8	7.6	2800+	2.5	8.1	2800+	2.0	6.9	2800+	3.9	7.9	2800+	1.6	5.6	2800+
1.7	0.8	2.5	2800+	1.0	4.0	2800+	0.9	3.7	2800+	0.9	2.8	2800+	0.8	3.3	2800+
1.8	0.6	2.0	2800+	0.5	2.3	2800+	0.4	2.4	2800+	0.7	2.4	2800+	0.5	3.1	2800+
1.9	0.6	2.7	2800+	0.2	1.2	2800+	0.3	2.0	2700	0.2	1.0	2800+	0.4	2.5	2800+
Sink 1.9	3.5	25.2	2200	1.5	12.6	2060	1.5	14.5	2260	4.0	31.4	2040	2.6	26.1	2120
	100.0	100.0		100.0	100.0		100.0	100.0		100.0	100.0		100.0	100.0	

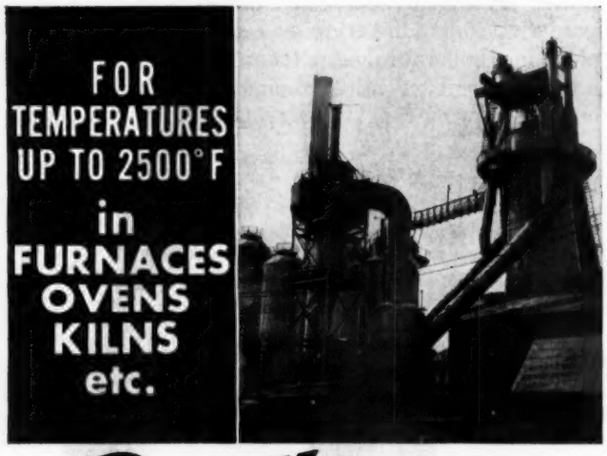
The free fusible matter occurs in a highly concentrated form, capable of becoming nuclei of numerous individual clinkers, with which the more refractory portions of the ash ultimately combine.

Fusibility of Ash in Separable Portions of Pulverized Coal

For this part of the investigation samples of the coals were pulverized to pass a 200-mesh screen and then separated into a number of gravity fractions from 1.3 to 1.9. The data, Table 3, show that, with the exception of the ash in the 1.3 float, the ash in all the other fractions down to the 1.9 sink was definitely of a highly refractory nature, so that there is a much sharper division between the fusible and the refractory than was found in the minus 4-mesh samples. This would indicate that, in the main, the ash in these coals is composed of two distinct classes of materials, one highly refractory and the other easily fusible, and that the two components of the ash are not only physically separable but are separated in the process of grinding.

From the standpoint of firing these coals in pulverized form, from 12.6 to 31.4 per cent of the total ash in the coal is fusible material introduced into the furnace in separate particles in a highly concentrated form, consisting of 60 to 70 per cent incombustible. The remainder of the ash, except a small portion found in the 1.3 float in some of the coals, is of a highly refractory nature.

The fusible material, being in separate particles, quickly melts in the furnace and adheres to the walls and tubes, forming a sticky film. A decreasing quantity of the fusible material is present in the gases as the distance from the burner increases. The refractory ash adheres to the film of fusible material in varying propor-



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Quality vs. Purity of Steam

By P. B. PLACE

Combustion Engineering Company, Inc.

The author discusses determinations of impurity in steam, the limitations of conductivity measurements, conflicting check tests for quality and purity, and cites the limits that should be specified under present methods of testing.

THE term "steam quality" has long been established to define the amount of moisture carried by saturated steam. It has no bearing on the amount of solids in the steam except as the latter may be inferred from a knowledge of the boiler water concentrations.

To avoid confusion and to define the amount of solids carried by steam, the term "impurity" has been used to refer to the solids content and the term "purity" has been employed to define the condition of the steam as regards solids content. A statement of the purity of steam is a statement of the amount of impurity that it contains and has no reference to moisture content except as mentioned above.

Quality is expressed as per cent of moisture in wet steam or as the per cent of dry steam in a wet mixture. It is determined by calorimetric methods which are of doubtful accuracy when the amount of moisture is less than 0.1 per cent. Expressing moisture values closer than the nearest 0.05 per cent is seldom justified.

The purity of steam is expressed by a statement of the amount of impurity in the steam. This amount is expressed as parts per million and should fundamentally be determined by gravimetric determination of evaporated residues. When the amount of impurity is less than 1 ppm very careful test methods are necessary to give accurate results and determinations of less than 0.5 ppm are of questionable accuracy. The errors involved are largely additive and it is easier to obtain results that are too high than it is to obtain results that are low. At best, evaporative methods are time-consuming and should be used chiefly for calibration of more rapid methods.

Limitations of Conductivity Measurements

The electrical conductivity or resistance of solutions of dissolved salts varies with the concentration and the determination of these characteristics may be made conveniently and continuously on a flowing sample. Conductivity tests determine only dissolved and dissociated salt solutions and do not register the presence of insoluble solids in the steam condensate. The latter, however, are normally such a small proportion of the total concentration in the boiler water that they are usually neglected.

Another limitation of the conductivity method is that some gases such as CO_2, NH_3 and H_2S, which may be present in the steam, dissolve in the condensed sample and contribute to its conductivity of the sample. Since, on evaporation, such gases do not yield a solid residue, they are not properly considered as an impurity that may deposit in the superheater or on turbine blades. Therefore, corrections must be applied to the observed conductivity results to reduce the latter to values that represent only solid impurity. These corrections are made in several ways but their accuracy is not completely established and methods are still in a stage of development.

The conversion of conductivity values into equivalent impurity values is a source of potential error in purity tests by the conductivity method. The conversion factor varies from 0.3 to 0.6, dependent on the composition of the impurity, a high proportion of caustic alkalinity giving a lower factor than a high proportion of neutral salts. It is evident that the accuracy of the reported impurity is directly proportional to the accuracy of the chosen factor. The factor should fundamentally be determined by gravimetric determination of evaporated residues but can be calculated if the analysis of the impurity is known. All of these limitations make conductivity tests of questionable accuracy when the impurity is below 0.5 ppm and in no case should such tests be reported closer than the nearest 0.1 ppm.

It should be expected that if both steam purity and steam quality tests are made on the same sample, the results of each test could be calculated in terms of the other if the concentration in the boiler water is known. Thus, if a boiler water has a concentration of 1500 ppm the moisture in the steam should have a similar concentration unless selective carryover occurs. Therefore, if steam purity tests show that the steam condensate has an impurity content of 3.0 ppm it may be assumed that the moisture which supplied this impurity was

$$\frac{3}{1500} \times 100 = 0.2 \text{ per cent.}$$

Similarly, if a steam quality test shows a moisture content of 0.2 per cent in steam generated from water having 1500 ppm it may be assumed that the impurity in the condensed steam, resulting from this moisture, will be

$$1500 \times 0.002 = 3.0 \text{ ppm.}$$

With concentrations of this order of magnitude, reasonable agreement between quality and purity determinations is obtained by careful testing. When, however, the boiler water concentration is low and the moisture carryover is also low it becomes difficult to check one method against the other because the accuracy of both calorimeter and conductivity tests decreases as the absolute values decrease. The errors involved are largely additive and one is more likely to obtain results that are too high than to obtain results that are too low.

Quality and Purity Tests Often Conflicting

Often, check tests on quality and purity give conflicting results that make one, or both, suspicious and in some cases absurd. For example, if from a boiler water having 150 ppm concentration, we get steam that tests 0.2 per cent moisture and at the same time gives impurity results indicating 0.9 ppm it is obvious that one or both results are in error. If the calorimeter result is correct the impurity should be down around 0.3 ppm, a value that is difficult to obtain with accuracy. Unless very careful impurity tests have been made and all corrections have been applied, it may be more probable that the 0.9 ppm is too high than that the 0.2 per cent is too low.

Conversely, if from boiler water having 3000 ppm concentration, we get steam testing 0.4 per cent moisture and impurity values indicating 3.0 ppm, it may be more probable that the impurity values are correct and the calorimeter value wrong, because it is relatively easy to get accurate impurity results at 3.0 ppm concentration.

The foregoing considerations are applicable to boilers that do not have steam washers. It is obvious that scrubbers which utilize the incoming feedwater to dilute and wash out the boiler water salts from the generated steam make any direct comparison between quality and purity impossible. Where no washer is used it may be assumed that the moisture in the steam is essentially boiler water having boiler water concentrations but where a washer is used, the final moisture in the steam is a mixture of both boiler water and feedwater. Since the concentration in these two constituents is considerably different and since the relative amounts of each are not accurately known, the actual concentration in the final moisture in the washed steam is uncertain. Therefore, tests of quality and purity on a washed steam are not as comparable as they are on an unwashed steam.

For example, from boiler water having 2000 ppm concentration, steam is generated and passed through a washer fed by feedwater having 200 ppm. There is not complete exchange in the washer between boiler and feedwater so that the moisture in the steam leaving the washer will be a mixture of boiler water having 2000 ppm and feedwater having 200 ppm. The relative amounts of each will depend on the efficiency of the washing and if it be assumed that this efficiency is 80 per cent, the concentration of salts in the final moisture may be estimated as follows:

$$2000 \times 0.20 = 400 \text{ ppm}$$
$$200 \times 0.80 = 160 \text{ ppm}$$
$$\text{Mixture} = 560 \text{ ppm}$$

Therefore, if the correct moisture content in the steam is 0.5 per cent, the impurity in the steam should be $560 \times 0.005 = 2.8$ ppm.

The actual efficiency of exchange in a washer is difficult to determine but independent estimates based on tests put it between 80 and 90 per cent. No doubt the efficiency varies with the type of boiler, type of washer, rating and similar factors. Though no direct comparison between quality and purity tests may be possible in boilers having steam washers, a number of tests under different operating conditions and in different plants will establish within reasonable limits an expected or normal efficiency of exchange in a washer which may be used in judging test results of similar equipment in other plants.

Very few, if any, tests on steam condensate show impurity contents of less than 0.2 ppm. When such low values are obtained they are more likely to be viewed with suspicion than to be accepted.

For example, consider a boiler with water concentration of 150 ppm using feedwater of 10 ppm for washing. An efficiency of 80 per cent for the washer gives a concentration of about 40 ppm in the final moisture in the washed steam. Although a test showing 0.4 ppm impurity might be considered good from a purity standpoint, this would represent over 1 per cent moisture, which would be regarded as a very poor quality.

Because of the difficulty of accurately determining small amounts of moisture and impurity in very pure condensate, it is not customary to guarantee performance values below fixed minimums even though better results are expected. For steam quality this low limit is 0.5 per cent moisture and for steam purity the low limit is 1 ppm of impurity. The present standards of testing may justify a change in these limits to 0.3 per cent moisture and to 0.5 ppm, respectively, in the future. At present, however, the reporting of moisture values of less than $^1/_2$ per cent, closer than the nearest 0.1 per cent, and of impurity values of less than 1 ppm closer than the nearest 0.1 ppm is not justified.

Correction—Due to a typographical error in the article on Slag Deposits in the February 1940 issue, page 38, middle of first column, should read "$^1/_8$-in. jet is more powerful."

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Combustion Calculations Using the Mol

By H. L. NORRIS, JR.

For those having to do with combustion calculations, use of the mol system is a time saver. This is explained and discussed with reference to the burning of coal, natural gas and oil, and convenient charts applicable to each of these fuels are included.

THE approach to boiler combustion calculations in a manner similar to the procedure used in internal combustion engine practice results in some simplification of the calculations. The simplification results from the expression of the quantities in mols, or, more specifically, in pound mols rather than in pounds.

A mol of a substance may be defined as the number of pounds of that substance numerically equal to its molecular weight. But even if the mol is defined as a number of pounds of a substance it also may be treated as a volume of that substance, with only the provision that the substance be in its gaseous stage. Thus it is a sort of common denominator between weights and volumes and it is here that its particular value is apparent.

Over a hundred years ago, Avogado said that equal volumes of different gases contained the same number of molecules just so long as their pressures and temperatures were the same. This gave rise to the conception of the mol. Under standard conditions of temperature and pressure of 32 F and 14.7 lb per sq in. abs. the mol occupies a volume of 359 cu ft. But for flue gas calculations all the products of combustion are at the same temperature because they are intimately mixed and the actual volume of a mol is of little importance. The significant thing is that the ratio of the number of mols of the different substances is also the ratio of their volumes.

As an example, consider fuel oil 13 per cent by weight hydrogen and 85 per cent by weight carbon, the remaining 2 per cent representing generally indeterminate amounts of sulphur, water, ash, etc. The needed data are:

	Symbol	Molecular weight	Atomic weight
Hydrogen	H_2	2	1
Oxygen	O_2	32	16
Nitrogen	N_2	28	14
Water	H_2O	18	..
Sulphur	S	..	32
Carbon dioxide	CO_2	44	..
Carbon monoxide	CO	28	..
Carbon	C	..	12
Air	...	28.94	..

Nitrogen-oxygen ratio by volume in air: 79.09/20.91 = 3.78.
Heat value of the oil: 18,600 Btu/lb.

Since the fuel oil is a mixture of a series of hydrocarbons, a chemical formula cannot correctly express it like the formula CH_4 expresses the gas methane. However, it can be expressed by a hypothetical chemical formula for the purpose of calculation. Of course, no chemical compound could exist having such a formula, but such hypothetical formula will enable simplified calculations to be made and the results will be the same as if a more tedious method were used.

Hypothetical Mol

The analysis of 85 per cent C and 13 per cent H by weight tells that for each 85 parts of carbon there are 13 parts of hydrogen. To convert this to parts by volume if it were in the gaseous state, divide the parts carbon and hydrogen by their respective atomic weights.

Thus 85/12 = 7.08 mols C and 13/1 = 13 mols H (not H_2). This is a hypothetical volume ratio of 7.08 volumes carbon and 13 volumes hydrogen. By dividing both 7.08 and 13 by 7.08 the ratio of 1 volume carbon to 1.835 volumes of hydrogen results. From this volume relation the hypothetical formula of $CH_{1.835}$ can be written to express the oil. It can be used just as if it were a true formula.

Equation for Combustion Reaction

Now assume combustion with only the required amount of air (zero excess) and that combustion is perfect.

$$CH_{1.835} + X\,O_2 + Y\,N_2 = A\,CO_2 + B\,H_2O + Y\,N_2 \quad (1)$$

To balance the equation all the carbon is burned to CO_2 and as there is one part of C in the left side of the equation there will be one part in the right side of the equation, therefore A is equal to 1. All the hydrogen is burned to H_2O and as there are 1.835 parts of H in the left side of the equation there will be 1.835 parts in the right side or 1.835/2 parts of H_2, therefore B is equal to 1.835/2 = 0.92. There are $(2A + B)/2$ parts of O_2, so $X = 1.46$. The ratio of nitrogen to oxygen in air is 3.78 and therefore the amount of N_2 on both sides of the equation is $3.78 \times 1.46 = 5.52 = Y$. Substituting the values of X, Y, A and B in equation (1) results in:

$$CH_{1.835} + 1.46\,O_2 + 5.52\,N_2 = 1\,CO_2 + 0.92\,H_2O + 5.52\,N_2 \quad (2)$$

The total number of mols in the right side of the equation is proportional to the total volume of the products of combustion and hence can be treated as the total volume of the products of combustion resulting from the combustion of one hypothetical mol of oil. The weight of oil represented is the total of the atomic weights of the elements of the oil multiplied by the respective amounts of those elements which is $12 \times 1 + 1 \times 1.835 =$ 13.835 lb.

The per cent by volume of the products of combustion will be expressed on two bases, wet and dry. The wet basis represents the condition that will exist if the fuel is

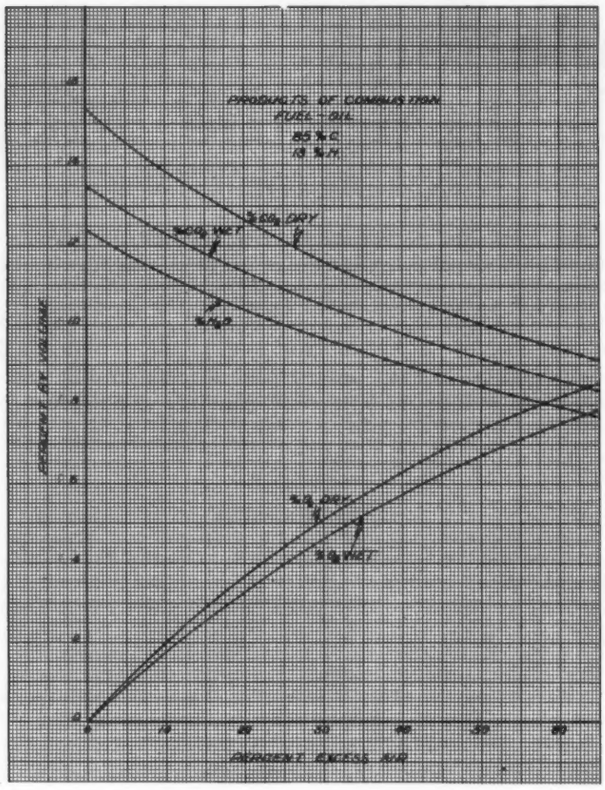

Fig. 1—Combustion chart for oil

completely burned with the required amount of air and the dry basis represents the condition that would exist if the water vapor were completely removed. The conventional method of expressing flue gas analysis is on the dry basis. The total volume considered in calculating the percentages by volume on a wet basis is the sum of the volumes on the right side of the equation. The total volume considered in calculating the percentages by volume of a dry basis is the sum of the volumes except H_2O of the right side of the equation.

PER CENT BY VOLUME

Wet		Dry	
CO_2	$1/7.44 \times 100 = 13.45\%$	CO_2	$1/6.52 \times 100 = 15.35\%$
N_2	$5.52/7.44 \times 100 = 74.3\%$	N_2	$5.52/6.52 \times 100 = 84.65\%$
H_2O	$0.92/7.44 \times 100 = 12.37\%$		

The results from the addition of excess air can be readily obtained by considering the right side of equation (2). Assume 10 per cent excess air. This means the addition of 10 per cent more of O_2 and N_2, both of which are unchanged by the combustion reaction and appear on the right side of the equation. As we have no further use for the left side of the equation it is omitted and the right side becomes:

$$1\ CO_2 + 0.92\ H_2O + 0.146\ O_2 + 6.07\ N_2 \qquad (3)$$

Total volume wet: 8.14 mols. Total volume dry: 7.22 mols.

PER CENT BY VOLUME WITH 10 PER CENT EXCESS AIR

Wet		Dry	
CO_2	$1/8.14 \times 100 = 12.3\%$	CO_2	$1/7.22 \times 100 = 13.85\%$
O_2	$0.146/8.14 \times 100 = 1.79\%$	O_2	$0.146/7.22 \times 100 = 2.02\%$
N_2	$6.07/8.14 \times 100 = 74.6\%$	N_2	$6.07/7.22 \times 100 = 84.0\%$
H_2O	$0.92/8.14 \times 100 = 11.3\%$		

Combustion of Gas

For another variation of this form of calculation consider a natural gas fuel consisting of 80 per cent methane (CH_4) and 20 per cent ethane (C_2H_6). If pure methane were burned the following would result if only the required amount of air were supplied:

$$CH_4 + 2\ O_2 + 7.57\ N_2 = 2\ H_2O + 1\ CO_2 + 7.57\ N_2 \qquad (4)$$

If pure ethane were burned the following would result if only the required amount of air were supplied:

$$C_2H_6 + 3.5\ O_2 + 13.25\ N_2 = 3\ H_2O + 2\ CO_2 + 13.25\ N_2 \qquad (5)$$

Since the gas is 0.8 methane and 0.2 ethane, multiply equation (4) by 0.8 and equation (5) by 0.2, obtaining:

$$\begin{array}{rl} & 0.8\ CH_4 + 1.6\ O_2 + 6.06\ N_2 = \\ 0.2\ C_2H_6 & \quad + 0.7\ O_2 + 2.65\ N_2 = \\ \hline 0.2\ C_2H_6 + 0.8\ CH_4 & + 2.3\ O_2 + 8.71\ N_2 = \end{array}$$

$$1.6\ H_2O + 0.8\ CO_2 + 6.06\ N_2 \qquad (6)$$
$$0.6\ H_2O + 0.4\ CO_2 + 2.65\ N_2 \qquad (7)$$
$$\overline{2.2\ H_2O + 1.2\ CO_2 + 8.71\ N_2} \qquad (8)$$

Equation (8) is the sum of (6) and (7) and represents the combustion of the assumed gas.

The percentages for each component of the flue gas for zero or any percentage excess air are calculated in the same way as for oil once equation (8) is established. By assuming different percentages of excess air the theoretical flue gas analysis for these different conditions may be quickly calculated and plotted.

Combustion of Coal

Consider the combustion of coal having the following ultimate analysis:

Carbon	79.86%	Nitrogen	1.86%	Moisture	2.9%
Hydrogen	5.02	Sulphur	1.18		14,351 Btu/lb
Oxygen	4.27	Ash	7.81		

For simplification, the per cent of sulphur is multiplied by the ratio of the atomic weights of carbon to sulfur (12/32) and added to the per cent of carbon. This gives the carbon equivalent of sulphur and it is carried through to show CO_2 as a product of combustion rather than SO_2. No appreciable error is introduced by handling the sulphur in this conventional manner.

Per cent carbon = $79.86 + (12/32 \times 1.18) = 80.30$
Per cent hydrogen 5.02 By weight

The hypothetical volume relationship is 80.3/12 carbon to 5.02/1 hydrogen = 6.69 to 5.02 or 1.33 to 1. The hypothetical formula corrected for sulphur will be $C_{1.33}H$.

Perfect combustion, no excess air:

$$C_{1.33}H + 1.58\ O_2 = 1.33\ CO_2 + 0.5\ H_2O \qquad (9)$$

$4.27/32 \times 5.02 = 0.027$ mol O_2 per mol $C_{1.33}H$ in the coal
$1.86/28 \times 5.02 = 0.013$ mol N_2 per mol $C_{1.33}H$ in the coal

Here the percentages of oxygen and nitrogen are divided by their respective molecular weights and the common divisor 5.02 is used in deriving the hypothetical formula. The required O_2 per mol $C_{1.33}H$ is $1.58 - 0.027 = 1.55$, and N_2 accompanying the required O_2, is $1.55 \times 3.78 = 5.85$. The required air is the sum of the O_2 and N_2 is $1.55 + 5.85 = 7.40$ mols air per mol coal. The required O_2 and its accompanying N_2 are to be used in calculating the per cent excess air.

The mol of coal is $12 \times 1.33 + 1 \times 1 = 16.97$ lb of combustible. Being 86.06 per cent combustible (C + H + S) and the mol is $16.97/86.06 = 19.7$ lb of dry coal or $19.7/97.1 = 20.3$ lb as fired.

Equation (9) is now rewritten:

$$C_{1.33}H + 1.58\ O_2 + 5.86\ N_2 = 1.33\ CO_2 + 0.5\ H_2O + 5.86\ N_2 \qquad (10)$$

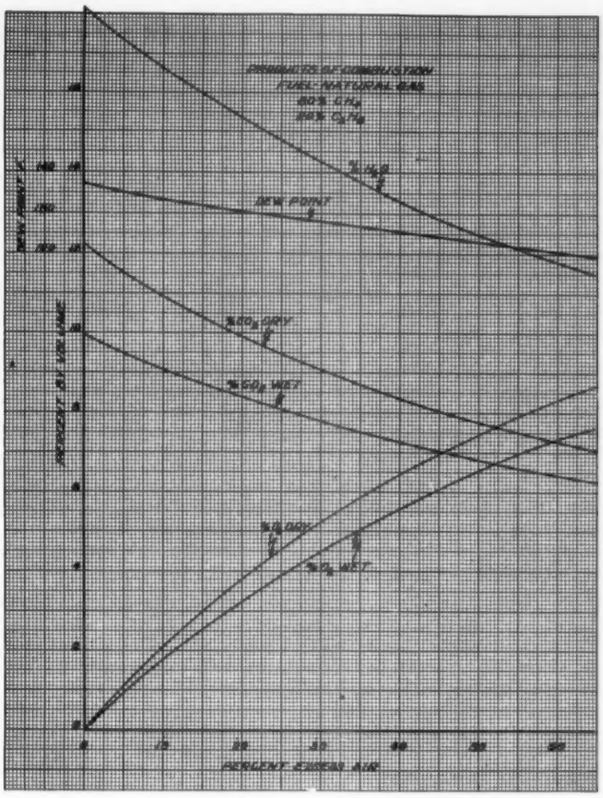

Fig. 2—Combustion chart for natural gas

The 5.86 mols N_2 is the sum of 5.85 mols accompanying the 1.55 mols O_2 and the 0.013 mol N_2 in the mol of coal.

By the use of equation (10) and varying the percentages of excess air assumed, the calculations are carried out in exactly the same manner as for the oil and gas fuel calculations above.

Figs. 1, 2 and 3 show the curves for the combustion of oil, gas and coal, respectively. In each case it has been assumed that combustion has been complete, all the hydrogen being burned to H_2O and all the carbon being burned to CO_2. For oil and gas with the proper distribution and mixing of the air and fuel, with the proper furnace temperature, the products of combustion should be given quite closely by the curves made up from theoretical calculations, with 10 per cent excess air or more. The same may be said of pulverized coal above 20 per cent excess air, but for the same coal being burned on a grate or on a stoker there is always the likelihood that the air distribution will not be just right or that the fuel bed will not be of proper or uniform thickness which will cause some of the carbon to be burned to CO rather than to CO_2.

Dew-Point

Dalton's law states in effect that the fraction of the total pressure exerted by a gas in a mixture of gases is its mol fraction. In the case of gas fuel with 10 per cent excess air the per cent of H_2O by volume is 16.65. The partial pressure exerted by H_2O is $0.1665 \times 14.7 = 2.45$ lb per sq in. abs. The steam tables give 133.8 as the saturation temperature which is the dew-point.

Ordinarily, the dew-point of the products of combustion has little or no significance because the usual stack temperature is very much above the dew-point. However, in some special instances where fuels high in sulphur or even containing free acids are being burned it is well to be certain that the dew-point is well below the exit temperature of the flue gases or rapid corrosion will result. If the per cent by volume of the wet gases has been calculated it is very simple to find the dew-point.

In the foregoing analysis, no account was taken of the amount of water present in the air supplied. For more accurate results, such as would be required in investigations of corrosion of air preheaters or steel stacks, account would have to be taken of the amount of water vapor present in the air supplied for combustion. The per cent by volume of water vapor in the air can be obtained from a psychrometric chart and is the mol fraction of water vapor present. This amount can be added to both sides of the equation expressing the combustion reaction.

Checking Orsat Analysis

Referring to Fig. 2, if an Orsat analysis showed 10.1 per cent CO_2, no CO and the per cent of oxygen was appreciably different from 3.4 the analysis would be known to be incorrect. Per cent excess air is the abscissa for both the CO_2 and O_2 curves and these quantities must check if the analysis is correct.

Density of Flue Gases

From the products of combustion expressed as the products of a reaction, the density of the flue gas is readily calculated. To find the density at 14.7 lb per sq in. and 300 F use $PV = wRT$. For a volume of one cubic foot, weight = density = PV/RT. Substituting $P = 14.7 \times 144$, $T = 300 + 460 = 760$ and $R = 1544 \times 13.2/367.77$, 13.2 is the total volume in mols of the products of combustion per mol of gas fuel, assuming 10 per cent excess air and 367.77 is the corresponding weight in

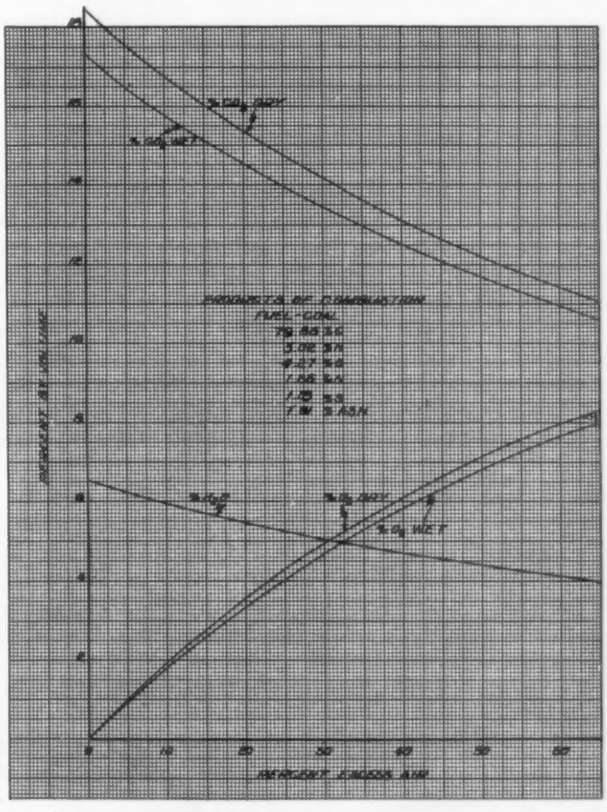

Fig. 3—Combustion chart for coal

pounds; 1544 is the universal gas constant.

$$\text{Density} = \frac{144 \times 14.7 \times 1 \times 367.77}{1544 \times 13.2 \times 760} = 0.0503 \text{ lb per cu ft.}$$

The volume of flue gas per 1000 Btu at 300 F and 14.7 lb per sq in. pressure, if the gas has a heat value of 1200 Btu per cu ft at 32 F and 14.7 lb, is found quite easily. The equation shows that for each volume of fuel gas, 13.2 volumes of flue gas result.

$$\frac{13.2 \times 760}{1.2 \times 460} = 18.16 \text{ cu ft of flue gas per 1000 Btu at 300 F}$$

Molecular Specific Heats

Since the computations have been carried out on the basis of mols of gas instead of pounds it is convenient to express the specific heats of the various gases on the per mol (MC_p) basis instead of on the per pound basis. These are generally accepted values for the range of from 0 to 600 F. Other values may be substituted merely by multiplying the specific heat per pound by the number of pounds per mol (molecular weight).

Gas	MC_p
Air, N_2, O_2 and CO	6.99
CO_2	9.82
H_2O	8.18
CH_4	10.39
C_2H_6	

To determine the specific heat for the flue gas in question it is necessary to multiply the specific heat of each component gas by its percentage in the mixture of flue gases. In the case of natural gas fuel and 10 per cent excess air the theoretical analysis of the flue gas is: H_2O, 16.65 per cent; N_2, 72.5 per cent; O_2, 1.74 per cent; CO_2, 9.1 per cent.

N_2 and O_2	0.742×6.99	5.180
CO_2	0.091×9.82	0.894
H_2O	0.1665×8.18	1.363
Molecular specific heat of mixture (flue gas)		7.437

Heat Balance

The items usually considered in a heat balance are: (1) heat absorbed by the steam generator; (2) loss due to moisture in fuel; (3) loss due to burning of hydrogen; (4) loss due to heat in dry chimney gases; (5) loss due to moisture in air; (6) loss due to incomplete combustion of carbon; (7) loss due to carbon in ash; (8) radiation and unaccounted-for losses.

The usual procedure is to express these quantities per pound of fuel and then convert to a per cent of the heat of combustion of the fuel. A more convenient way when the calculations have been carried out as outlined above is to express all quantities per mol of fuel. For this purpose, the mol of gas fuel has to be explicitly defined. This is done by stating the pressure and temperature of the gas fuel. Usually, the standard conditions to which gas fuels are referred to are 32 F and 14.7 lb per sq in. As previously pointed out, under these conditions a mol of gas occupies 359 cu ft. Solid or liquid fuels can be expressed as mols by dividing the weight of fuel in pounds by its molecular weight.

The items of the heat balance will now be discussed individually:

(1) The heat absorbed by the steam generator. This is evaluated by measuring the amount and condition of the steam produced.

(2) Loss due to moisture in fuel. Usually, this may be neglected for oil fuel, but for gas or solid fuel it may or may not be taken into account depending on the amount of moisture as shown by the analysis. In the case of solid fuel it is the heat required to evaporate the water from atmospheric pressure and superheat it to stack temperature. In the case of gas fuel, only the sensible heat added in raising the temperature from fuel temperature to stack-temperature is lost because the water is already evaporated before burning the gas.

(3) Loss due to the burning of hydrogen. This loss is readily evaluated. The equation for combustion, before being corrected for the water vapor in the air, shows on the right side the mols of water per mol of fuel. The heat quantity is the latent and sensible heat required to evaporate the water and superheat it to stack temperature. The latent heat per mol of water at atmospheric pressure is $970.2 \times 18 = 17{,}463.6$ Btu. For the case of natural gas and a fuel temperature of 70 F and a stack temperature of 350 F this loss amounts to:

$$2.2[(212 - 70) \times 18 + 17{,}463.6 + (350 - 212) \times 8.18] = 46{,}540 \text{ Btu per mol of fuel}$$

The heat content of the fuel being 430,800 Btu per mol, the per cent loss due to the burning of hydrogen is $46{,}540/430{,}800 \times 100 = 10.82$ per cent.

(4) Loss due to heat in dry chimney gases. The heat lost due to the water from the burning of hydrogen to steam has been completely evaluated under item (3) so this item will consider only the heat loss by the remaining portions of the flue gases. The molecular specific heat of the flue gas resulting from burning of gas fuel with 10 per cent excess air is 7.3 (dry products only). The amount of the dry products of combustion per mol of fuel gas is the sum of all the products except H_2O which is 11 mols. The heat loss then is:

$$11 \times 7.3(350 - 70) = 22{,}500 \text{ Btu per mol of fuel gas}$$
$$22{,}500/430{,}800 \times 100 = 5.52\%$$

(5) Loss due to moisture in air. Assume that the temperature and relative humidity of the air are such that the air is 1.13 per cent water vapor. For gas fuel and 10 per cent excess air there are supplied $(2.3\ O_2 + 8.7\ N_2)\ 1.10 = 12.1$ mols of air per mol of gas fuel. Let X equal the mols of H_2O accompanying this air. $(12.1 + X)\ 0.0113 = X$. $X = 0.138$ mol H_2O per mol gas fuel. The heat loss is:

$$0.138 \times 8.18\ (350 - 70) = 317 \text{ Btu per mol of fuel}$$
$$317/430{,}800 \times 100 = 0.074\%$$

(6) Loss due to the incomplete combustion of carbon. Usually, there is no more than a trace of CO when burning natural gas with 10 per cent excess air. Assuming that the flue gas analysis showed 0.5 per cent CO there would be $0.005 \times 13.2 = 0.066$ mol of CO per mol of fuel gas burned. A mol of CO represents a heat loss of 122,640 Btu.

$$0.066 \times 122{,}640 = 8{,}100 \text{ Btu per mol of fuel gas}$$
$$8{,}100/430{,}800 \times 100 = 1.88\%$$

(7) Loss due to carbon in ash. This item disappears when burning gas or oil fuel. For solid fuel, the molecular units used in calculation offer no advantage over the conventional method of calculating.

(8) Radiation and unaccounted-for losses. This item is taken by the difference between 100 per cent and the total of items 1 to 7, inclusive.

REVIEW OF NEW BOOKS

Any of the books reviewed on this page may be secured from Combustion Publishing Company, Inc., 200 Madison Ave., New York

The Mathematical Theory of Non-Uniform Gases

By Sydney Chapman and T. G. Cowlings

The book is written for the trained mathematician, physicist or physical chemist who is familiar with the kinetic theory of gases rather than for the engineer. A short chapter on the vector and tensor notation is put at the beginning. Another chapter entitled "Definitions and Theorems" shows how the tensor calculus is applied to the specific subject of gases. The treatment considers mixtures of gases as well as simple gases. The usual theory of velocity distribution according to Maxwell, Boltzmann's H-theorem and Jean's theory of the persistence of velocities form the basis of the text. The kinetic theory is discussed from the viewpoint of the rigid spherical molecule, molecules which are centers of force, molecules which have both attractive and repulsive fields of force and molecules which possess energy of rotation. The latter are called rough spherical molecules. The authors discuss the phenomena of momentum transfer such as viscosity, thermal conduction and diffusion for the various molecules and gas mixtures. Here, the work of Enskog has been followed. More recent subjects such as dense gases, the quantum theory as applied to gases and the behavior of gases in ionized fields are subjects of discussion.

It is to be regretted that the treatment is so highly mathematical that in many places it obscures the physical background. The book can be used as a reference since the kinetic theory of gases is given a more general and accurate treatment than is usual.

The price of the book is $7.50.

Statistical Thermodynamics

By R. H. Fowler and E. A. Guggenheim

Written for the physicist this book is of value to one concerned with research on the thermodynamic properties of matter. The entire development of statistical mechanics from classical to recent conceptions and the experimental thermodynamic results up to present times have been assembled by the authors into a brilliant representation of thermodynamics from the statistical point of view. In the endeavor fully to explain the reasoning underlying the statistical method, the authors commence with a mathematical derivation of statistical laws in the most general form and then introduce the various restrictions leading to the specialized systems of matter. Therefore, the reader who wants information on a special subject is unable to understand the respective section without the prior study of the abstract introduction.

The main objective of statistical thermodynamics is to set up the partition function for the various systems of matter. Then, the thermal properties are derived from this function by a purely thermodynamic procedure.

The first system discussed is the perfect gas. Here, Maxwell's distribution law is derived to serve as a basis for the time-dependent phenomena in a later section on Chemical Kinetics, and the partition function of the different molecule models is given for classical and quantitative energy distribution.

In the chapter on imperfect gases, potential energy is introduced into the partition function, the ensuing thermodynamic relations are derived, and the statistical implications of some empirical equations of state are analyzed. Fermi statistics are used in the investigation of the properties of the electron gas. From the statistics of surface layers, a general theory is derived for the phenomena first investigated by Langmuir.

Solid bodies provide an ample field for application; the theory of crystals is based on Debye's approximative laws, and is extended, in a comprehensive separate chapter, to the general theory of imperfect crystals, crystal lattices, and solutions of crystals. The investigation of chemical equilibria and vapor pressures leads to the statistical interpretation of Nernst's law.

Further chapters are devoted to the theory of liquids, solutions of non-electrolytic and electrolytic liquids and to electrical properties of matter. Numerous experimental results from all fields illustrate the theoretical expositions.

The price of the volume is $9.50.

A History of the Growth of the Steam Engine

By Dr. Robert H. Thurston

This is a reprint of Professor Thurston's classic treatise on the steam engine, originally written in 1878 and subsequently revised through several editions prior to and shortly after the author's death in 1903. The present edition has been brought up to date by a supplementary chapter prepared by W. N. Barnard, Director of the Sibley School of Mechanical Engineering, Cornell University.

In this 550-page book, with 182 illustrations, the story of the steam engine, from its beginning down to modern applications is told in a most fascinating manner. Despite the fact that the steam turbine has supplanted the steam engine in many fields, industry and social progress owe much to the latter. In this connection, the text links the steam engine with industrial progress and portrays the contributions of numerous pioneers in its application to the stationary, railway and marine fields. Aside from providing delightful reading, it serves as an informative reference.

Handsomely bound in cloth with black and copper stampings the book sells for $3.

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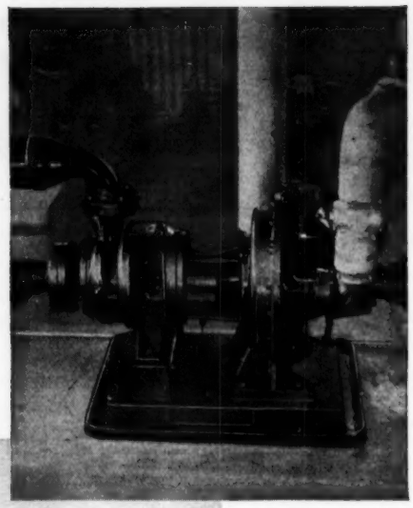
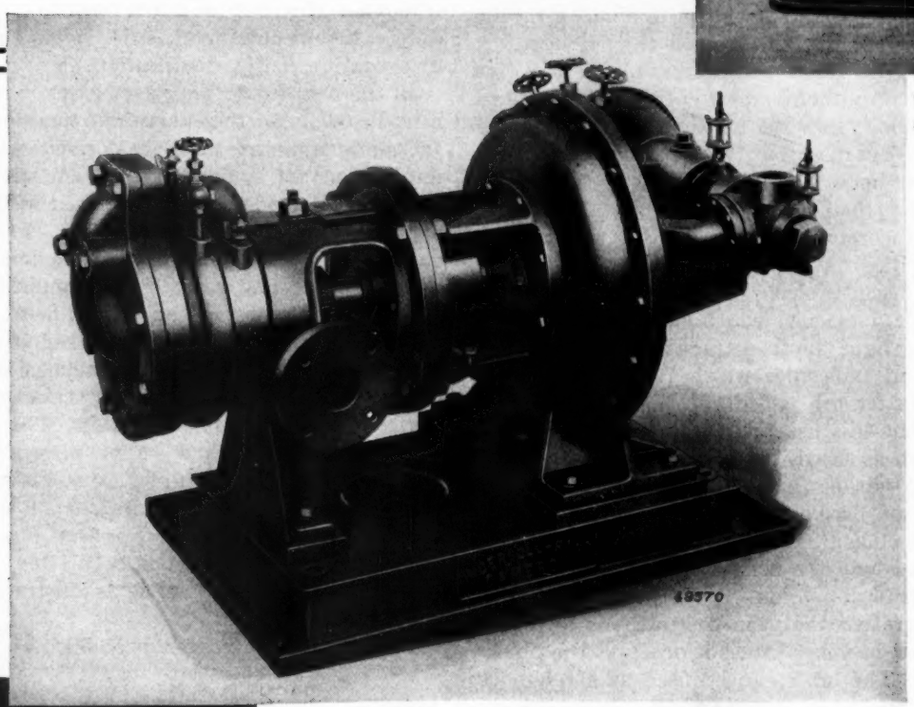
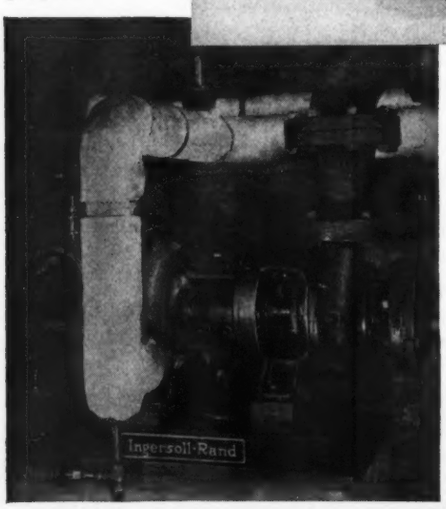


Turbine pump installed in a refinery.



Two-stage turbine-driven pump.

Turbine pump in a gas plant.



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STEAM ENGINEERING ABROAD

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High-Speed Turbine with Special Double-Casing Construction

In *VDI Zeitschrift* for October 1939 Dr.-Ing. Karl Roeder describes a high-speed, relatively high-pressure turbine built for and tested at the Technische Hochschule at Hannover, Germany. The unit is rated at 1000 kw and operates at 16,000 rpm with 654 lb initial pressure, 806 F total steam temperature and 18.5 lb back pressure. Its special features include a double-shell construction of unusual design.

The author points out that the problem of designing a steam turbine for effectively handling small volumes of steam becomes more significant with increase in pressure, because the efficiency of the high-pressure stages is substantially lower than that of the intermediate and lower stages. For a narrow range of the steam band the rim losses become large and exert the greatest influence at low loads. To reduce these losses a high rotative speed was adopted so that, for equal rim speeds, the stage diameter becomes smaller. Since the heat drop and the steam velocity remain the same the blade lengths increase, giving a wider steam band, and the relative rim loss becomes smaller. Small clearances between the rotating and stationary parts are necessary, particularly for the reaction blading, and the construction must be such as to maintain the small clearances.

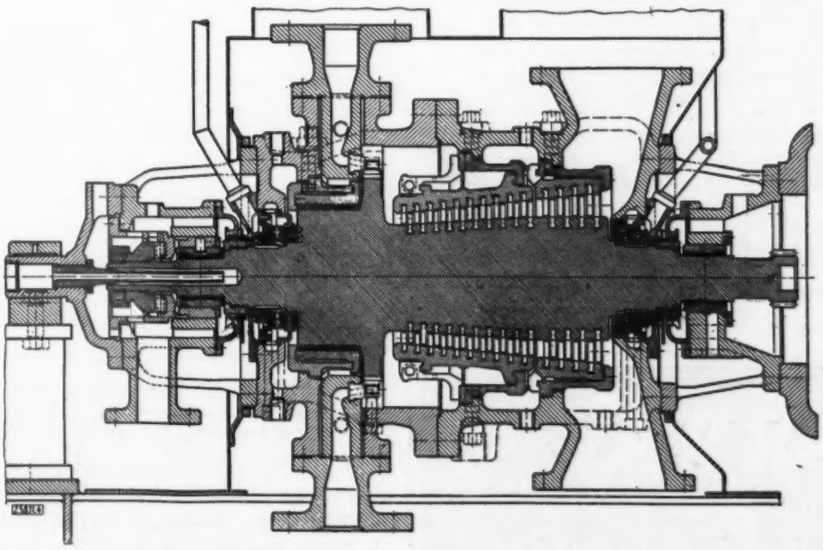
To avoid the effects of casing distortion and consequent variation in clearance, double-shell construction was employed in this unit as shown in the cross-section. The inner shell which carries the stationary blades is cylindrical and attached to the turbine housing in such a manner as to take up the thrust yet remain circular in shape. The inner shell follows temperature changes of the steam more quickly than do the rotating parts, hence the radial clearances are somewhat greater in starting, although during operation this difference in temperature is only about one-third to one-fourth of that during starting. Hence, the clearances may be made smaller than in the conventional turbine in which the temperature of the fixed clearance boundary pieces lag behind the steam temperature more than does the runner.

This turbine has one velocity and fourteen reaction stages. Governing is by throttling, but since the first is a velocity stage it may also have nozzle-orifice regulation with the two nozzles 180 deg apart. The blades are not shrouded but the free ends are bevelled in the usual manner. The housing has radial stuffing boxes.

The weight of the unit is given as 0.62 kg (1.37 lb) per kw, not including the governor nor the gears through which the generator is driven. On test the water rate at a load of 974 kw was 9.88 lb per kwhr.

Power Station Trends in England

Sir Leonard Pearce, Chief Engineer of the London Power Company, in his recent Hawksley Lecture before the Institution of Mechanical Engineers, reported in *Electrical Review* of January 26, traced the progress of steam power generation over a number of years and reviewed present trends, particularly practice in Great Britain. Expressing the opinion that superposition is not likely to become popular in England, he sensed a



Cross-section through 1000-kw, 1600-rpm turbine showing cylinder construction

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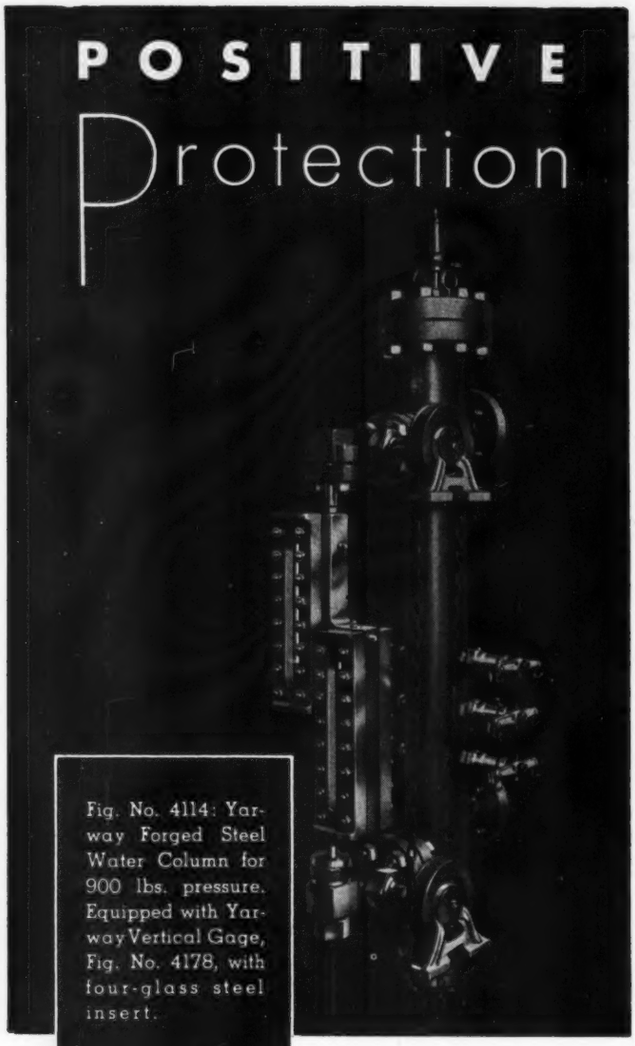

Fig. No. 4114: Yarway Forged Steel Water Column for 900 lbs. pressure. Equipped with Yarway Vertical Gage, Fig. No. 4178, with four-glass steel insert.

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trend toward decentralization through the construction of more stations of moderate size, around 250,000 kw, despite possible reduction in capital costs and considerations of efficiency offered by larger stations. There also appears to be a definite limit to the use of turbine-generators of 75,000 and 100,000 kw in large base-load stations in that country. There are at present in service or under construction eighty machines of 25,000 kw at 3000 rpm.

Of thirty large natural-circulation boilers installed during the last three years, two were of 550,000 lb per hr capacity, 1420 lb per sq in. pressure and 965 F total steam temperature and twenty-eight ranged in capacities of 300,000 to 375,000 lb per hr. Of these, twelve were for 375 lb pressure, 730 to 780 F total steam temperature, and sixteen of 600 to 675 lb pressure and steam temperatures ranging up to 910 F. Ten are fired with pulverized coal, six have chain grates and fourteen, including the two largest units, have multiple-retort underfeed stokers.

While conceding that steam temperatures of the order of 1000 F might eventually be adopted, Mr. Pearce did not consider that much was to be gained by exceeding 1500 lb pressure.

Commenting on forced-circulation boilers, the author expressed the opinion that their advantages, particularly as to space requirements, must be offset by more elaborate controls and power required to drive the circulating pumps. Those installed or under construction in England include fifteen of the LaMont type, five Sulzer, four Loeffler and one Velox.

A New Combustion Turbine

Escher Wyss has developed a combustion turbine operating on a new principle, which is described by F. Liceni in *VDI Zeitschrift* of November 25, 1939. It involves a closed circuit in which the products of combustion do not flow through the turbine but, instead, heat air as a working medium which passes through a separate circuit, as indicated in the diagram, Fig. 1. The turbine is therefore kept free from deposits.

The air which is heated by the products of combustion in the gas heater L expands through the turbine from

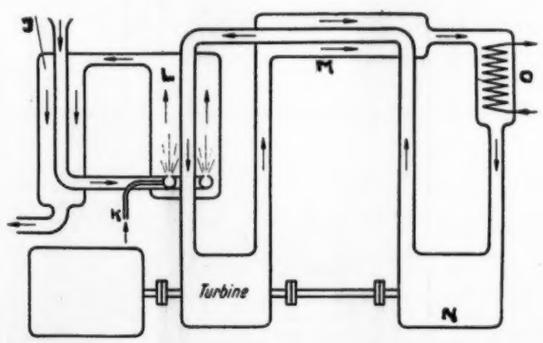


Fig. 1—Diagram of Escher Wyss combustion turbine circuit
J is preheater for combustion air, *K* the fuel supply, *L* the gas heater, *M* the heat exchanger, *N* the compressor and *O* the aftercooler.

pressure p_1 to p_2, the back pressure, p_2 being greater than atmospheric. It then flows through a heat exchanger M, in which it gives up heat to the air leaving the compressor N, and then passes on to an aftercooler O where its temperature is further lowered before reaching the

suction of the compressor. Air for combustion is preheated by the flue gases of the gas heater at *J*.

The cycle is indicated in Fig. 2. Expansion occurs along the line *AB*. Heat transfer in the heat exchanger is represented by *BC* and *DE*, and the air is isothermally compressed along *CD*. In the actual machine compression occurs in several stages with intercooling. The products of combustion in the gas heater heat the air along the line *EA*.

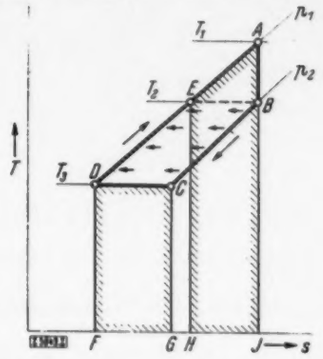


Fig. 2—Operating cycle diagram

The theoretically available work is indicated by the area *ABCD* and is equal to the difference between the work done by the turbine and that done by the compressor. Work done by the turbine is equal to *AJHE*, the outside heat supplied, while that done in compression equals the heat carried away by the compressor cooling water, as represented by area *DCGF*.

The author then shows, theoretically, that for air taken as a perfect gas, and with T_1 and T_3 given predetermined values, the efficiency of the unit, neglecting losses, will depend only upon the pressure ratio of p_1 to p_2. The smaller this ratio, the higher will be the efficiency.

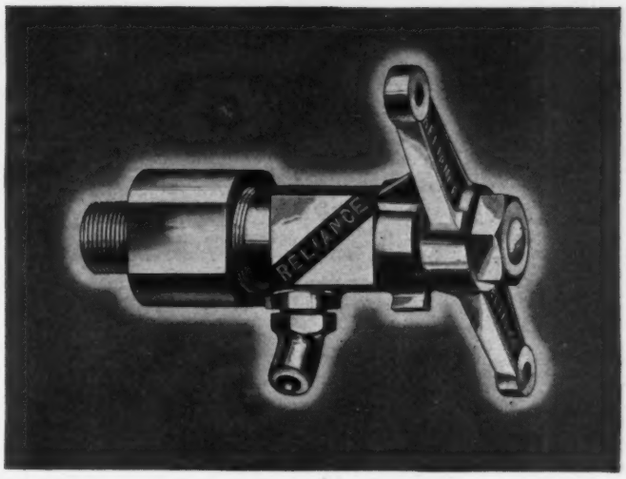
Should the unit operate with an open exhaust, as is the case with the Brown Boveri combustion turbine, a very large heating surface would be required for the heat exchanger. If, however, the closed cycle operates under a back pressure, smaller surfaces may be employed; also, both the turbine and the compressor become smaller.

The turbine load may be regulated by varying the back pressure and thereby the pressure in the circuit. If the back pressure be lowered from 9 to 1 atmosphere by bleeding off a portion of the circulating air the load will be lowered by one-tenth. At constant revolutions the efficiency of the unit will remain nearly constant.

W. J. Vogel Promoted

The appointment of W. J. Vogel as Assistant Chief Engineer of Combustion Engineering Company, Inc. has been announced by John Van Brunt, Vice President in charge of engineering.

After receiving his technical education at Rensselaer Polytechnic Institute and Brooklyn Polytechnic Institute, Mr. Vogel spent some time in charge of the power plant at the Passaic Print Works and later with the Babcock & Wilcox Company. He entered the employ of Combustion Engineering Company in 1924 as a contract engineer. Since then he has spent four years in engineering sales work in the Philadelphia and Detroit offices of the company and most of the remaining time as a contract engineer in the Engineering Department.



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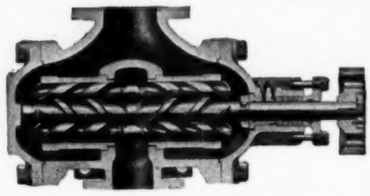

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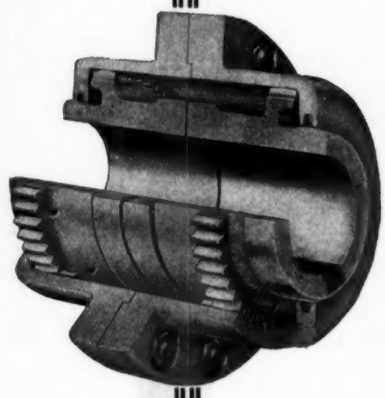

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